1978

A thermodynamic study of a representative gas turbine for design of a computer module for steady state engine monitoring and trending.

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http://hdl.handle.net/10945/18565

A THERMODYNAMIC STUDY OF A REPRESENTATIVE GAS TURBINE FOR DESIGN OF A COMPUTER MODULE FOR STEADY STATE ENGINE MONITORING AND TRENDING.

Thomas George Tetlow

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE

READ INSTRUCTIONS BEFORE COMPLETING FORM

1. REPORT NUMBER	2. GOVT ACCESSION NO.	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) A THERMODYNAMIC STUDY OF A REPRESENTATIVE GAS TURBINE FOR DESIGN OF A COMPUTER MODULE FOR STEADY STATE ENGINE MONITORING AND TRENDING.		5. TYPE OF REPORT & PERIOD COVERED THESIS 6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) TETLOW, THOMAS GEORGE		8. CONTRACT OR GRANT NUMBER(s)
9. PERFORMING ORGANIZATION NAME AND ADDRESS MASS. INST. OF TECHNOLOGY		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS
11. CONTROLLING OFFICE NAME AND ADDRESS CODE 031 NAVAL POSTGRADUATE SCHOOL MONTEREY, CALIFORNIA, 93940		12. REPORT DATE MAY 78 13. NUMBER OF PAGES 133
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office)		15. SECURITY CLASS. (of this report) UNCLASS 15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

APPROVED FOR PUBLIC RELEASE; DISTRIBUTION UNLIMITED

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

THERMODYNAMIC STUDY, GAS TURBINE, ENGINE MONITORING & TRENDING

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

SEE REVERSE

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by

THOMAS GEORGE TETLOW

B.S., United States Naval Academy
(1972)

SUBMITTED IN PARTIAL FULFILLMENT

OF THE REQUIREMENTS FOR

THE DEGREE OF

MASTER OF SCIENCE IN OCEAN ENGINEERING

AND THE DEGREE OF

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

at the

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

May, 1978

A THERMODYNAMIC STUDY OF A REPRESENTATIVE GAS TURBINE FOR DESIGN OF A COMPUTER MODULE FOR STEADY STATE ENGINE MONITORING AND TRENDING

by

THOMAS GEORGE TETLOW

Submitted to the Department of Ocean Engineering on May 12, 1978 in partial fulfillment of the requirements for the Degree of Master of Science in Ocean Engineering and for the Degree of Master of Science in Mechanical Engineering.

ABSTRACT

The purpose of this paper is to develop equations which represent the steady state operation of the LM2500 marine gas turbine engines and which could be integrated into an inexpensive computer for use in trending the engine. In the first three chapters, the topic will be introduced and the background discussed briefly. Then a maintenance plan, which will utilize a small computer module and will yield results identical to a large complex system, will be presented. The fourth chapter of the paper presents a discussion on free power turbine theory. The conclusions of the chapter are (1) that the operation of the gas generator is restricted to a line on the compressor characteristics, and (2) the pressure and temperature ratios across the high pressure turbine are constant when the power turbine is choked. The result of the second conclusion is that the output of the power turbine is essentially independent of compressor speed. Chapter V discusses the trending procedure and establishes the parameters necessary to ensure an informative system. It is also established that the parameters can be represented as a function of gas generator RPM as a result of the conclusion drawn from Chapter IV. In Chapter VI, the operational baseline is established. It is accomplished graphically by establishing a gas generator RPM which corresponds to minimum SFC (the data used is from the LM2500 performance data), and the RPM then is used to determine other engine parameters. In the final stage, the data is refined using the theory

developed in Chapter IV. The results of Chapter VI are the six monitored parameters expressed as a function of gas generator speed. In the next chapter, a procedure for applying corrections is discussed and the logic presented. The corrections enable the trending system to be useful over varying ambient conditions. And finally in Chapter VIII, the complete equations representing the deviations and incorporating corrections are assimilated. This is the primary result of Chapter VIII. But in addition, other uses for the equations are discussed, and the idea of a new design for a control system is presented.

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Title: Associate Professor of Mechanical Engineering

Reader: A. Douglas Carmichael

Title: Professor of Power Engineering

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CHAPTER I

INTRODUCTION

With the concern over the shortage of fossil fuel today, various systems are being developed to minimize costs and fuel consumption. Some of these are electronic controls, on-condition monitoring systems, and data trending programs. The General Electric marine gas turbine engine, the LM2500; installed in the DD963 Spruance class destroyers; does not have a system to perform on-condition monitoring or trending. However, a trending system could be developed for the LM2500 which is simple and inexpensive. The basis of such a system is to compare actual engine conditions to a baseline, which is the conditions of a new, finely tuned engine. The deviations, when plotted against time, will indicate trends. The major component would be a small analog-digital computer module which would be replaceable as a complete unit. The module would have the baseline programmed into the circuitry which would enable it to calculate the deviations when the actual parameters were fed in. The trending program would enable the engineer to determine when the engine is operating inefficiently and then take corrective action. For example, in a gas turbine, a one percent reduction in compressor efficiency results in a 1-1/2% increase in fuel flow. The system

would detect the problem when the deviation was only one or two percent, whereas normally, it would not be detected until the deviation was three to five percent. If maximum deviation limits were established, the engineering officer would be able to predict the time of component or engine replacement. In addition to fuel savings and decreased maintenance costs then, the availability of the ship for extended deployments would increase, and scheduling would be less susceptible to changes. The overall goal of a trending system is to affect repairs or replacement of an engine when the deviation dictates, rather than when the fixed overhaul life cycle expires.

This paper will develop the operational baseline for the LM2500 as a function of gas generator speed, thereby providing the basis for the computer module and other possible applications. The baseline operation will be determined from Reference [1], which is a listing of component conditions for various operating regimes, and it will be established for minimum fuel flow at steady cruising speeds. It will enable the engineer to trend the engine at any time, independent of speed. It would not involve the complexity of a large computer system. Therefore, it would be operational a larger percent of the time and would not involve electronic technicians nor additional

training for engineering personnel. And finally, it could perform a similar function as reliably as an expensive, complex computer system.

CHAPTER II

BACKGROUND

Trending gas turbine engines in general is not a new concept. Airlines have established several methods of monitoring the performance of jet engines. At first, these methods involved recording data while in flight and feeding it into a computer once the plane had landed. This system has evolved to on-board monitoring utilizing installed computers. On-condition monitoring systems have been built and installed in a few industrial gas turbine power plants, but generally, trending the land based plants is a recent development. These sytems can be extremely expensive and complex. They involve large computers with man-machine interfaces. The printout tells the engineer component efficiencies and specific degradations. In addition, it determines maintenance actions required, and the engineer then tells the computer when the maintenance action has been accomplished. These gas turbines operate generally at constant speed, at their design point. The marine gas turbine, however, cannot be considered a constant speed machine. It operates continually off-design. The concept of trending marine gas turbines is relatively new, but several vessels have complex systems installed such as those discussed above. And presently, many engines are being

purchased with the monitoring equipment built into the package. However, for most gas turbines existing now without the monitoring equipment, various trending programs are being applied. Most of these programs are accomplished by hand. They involve reading gauges, applying corrections, and then comparing the corrected value to a baseline curve supplied by the manufacturer. The Canadian and British Navies employ this procedure. The calculations are done by hand and then compared. In the Canadian Navy, the baseline operations are plotted against pressure ratio of the compressor, and the British Navy uses the compressor speed. However, the programs are very similar. The United States Navy does not perform trending on their gas turbine engines.

Trending the engines is a reliable method to detect ensuing problems. However, it should not be used as the only method of monitoring engine performance. Other methods and equipment that are helpful are shown in the list below [3].

- Visual Inspection
- Borescoping
- Vibration Analysis
- Sonic Analysis
- Spectrometric Oil Analysis (SOA)
- Magnetic Chip Detection

- Filter Analysis
- Particle Counters
- Eddy Current Meters
- Radiography
- Ultrasonics
- Strain Gaging
- Monitoring During Start-Up and Shut-Down

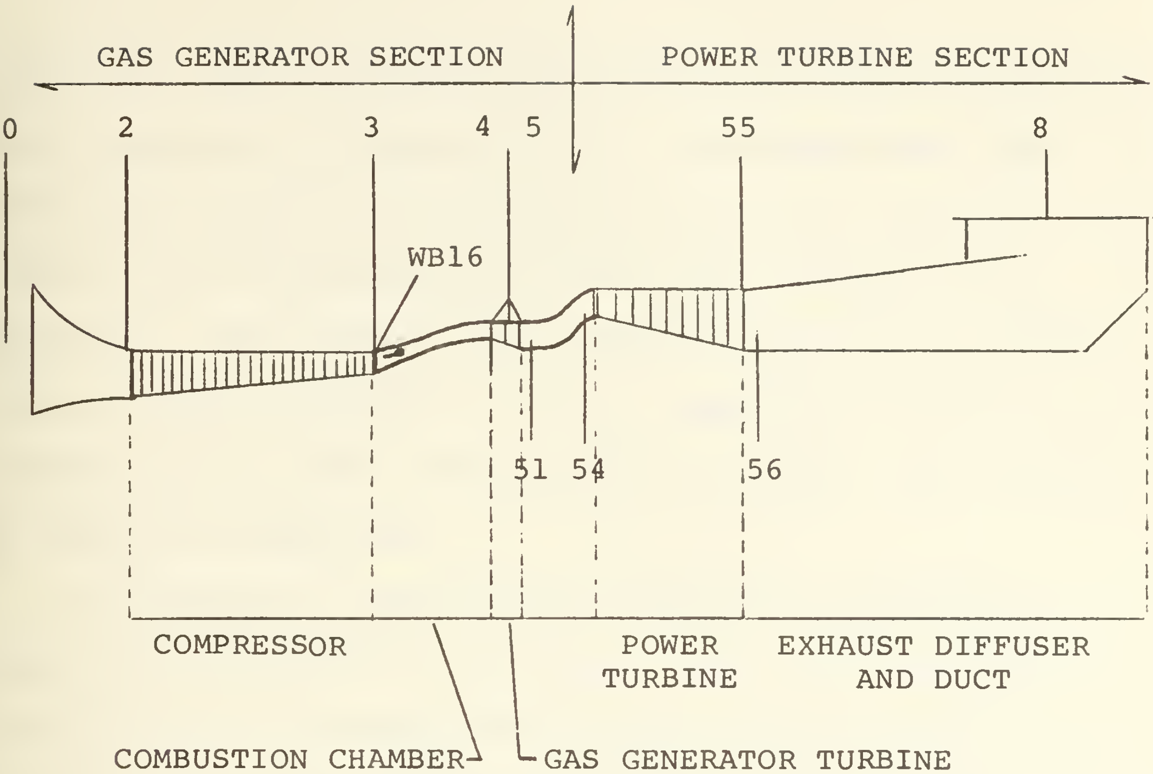
Of these methods, several can be eliminated from ship's force applications. Sonic analysis and SOA are costly and require additional expertise and training. Particle counters and eddy current meters are in a new stage of development and therefore not reliable. Radiography, ultrasonics, and strain gaging are complex, costly, and require special technicians. The methods chosen by the marine engineer to monitor the engine must yield useful information, be easily applied, and be cost effective in the marine application. The expensive and precise equipment should be limited to use by repair facilities at periodic intervals; thereby reinforcing the ship's program. The selection from the list above, when combined with a reliable trending program, should enable the engineer to evaluate the engine continually for a minimum cost outlay; thus performing the identical function as the large computer system.

CHAPTER III

THE LM2500

The LM2500 gas turbine is a free power turbine engine. The gas generator is a simple cycle, axial flow, gas turbine which consists of a single compressor with non-variable stator vanes, combustor, and a high pressure turbine. The exhaust from the high pressure turbine is directed through the free power turbine which is mechanically geared to the ship's propellor. There is no mechanical connection between the two turbines, but the compressor and high pressure turbine are on the same shaft. A schematic of the engine, which also specifies the station numbers, is shown in Figure 3-1. The station numbers designate where the readings of temperature and pressure are taken. An explanation of the nomenclature can be found in the appendix. The design horsepower rating of the engine is 21500 (with four to six inches of water inlet and exhaust duct losses), but the engine can attain upwards of 28000 horsepower depending on ambient conditions. General values of engine parameters can be found in the appendix.

It should be noted that most of the measurements taken on the machine are total conditions, but a few are static. The theory developed in the next few chapters is for use with total conditions, as designated by the "zero" in the



STATION		LOCATION
0		AMBIENT
2		COMPRESSOR INLET
3		COMPRESSOR DISCHARGE
4		G G TURBINE INLET
5		G G TURBINE DISCHARGE (WITHOUT COOLING FLOW)
51		G G TURBINE DISCHARGE (WITH COOLING FLOW)
54		P T TURBINE INLET
55		P T TURBINE DISCHARGE (WITHOUT COOLING FLOW)
56		P T TURBINE DISCHARGE (WITH COOLING FLOW)
8		EXHAUST DUCT FLANGE
P	=	PRESSURE
T	=	TEMPERATURE
W	=	AIR OR GAS FLOW
WF	=	FUEL FLOW
WB	=	BLEED AIR FLOW

FIGURE 3-1: LM2500 Station Number Locations [1]

subscript. For the trending program, the difference between total and static is ignored. It is considered that the additional change in the parameter due to the change in velocity has a negligible effect on the outcome of the procedure. The purpose of a trending program is to detect changes, and for trending it is not important that the pressure ratio across the compressor be exact, but instead it should be consistent.

In Chapter IV, the station numbering is modified to a basic gas turbine numbering system. This is illustrated in Figure 3-2. The following chapters utilize the system in Figure 3-1, which is standard for the LM2500. The stations in Figure 3-1 correlate to the actual positions of measurements of the conditions whereas Figure 3-2 represents the general theory.

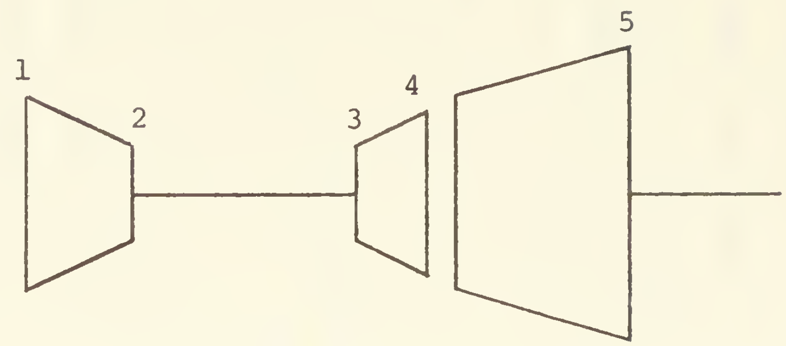


FIGURE 3-2: General Gas Turbine Station Numbers

CHAPTER IV

OFF DESIGN OPERATION

The variable speed operation of a gas turbine is quite different from the constant speed case. Turbines and compressors are designed to operate at one point. At this point, the design point efficiency is usually maximum because the fluid flow over the blades is optimum; the induced incidence is such that flow separation is minimized; and maximum enthalpy change across the blade is obtained. Therefore, when a turbine or compressor operates off design, and this is particularly critical with the compressor, the work is not maximized, and the most notable effect is the rapid increase in specific fuel consumption. However, these basic rules apply mainly to the simple cycle, but these unsatisfactory conditions can be improved considerably with a more complex cycle. Each component, whether turbine or compressor, has a unique characteristic chart referred to as "compressor characteristics" or "turbine characteristics". The operational range of these charts for the compressor can be large. But when a turbine is coupled with the compressor, the operating range of the compressor becomes a smaller zone. And when a free turbine is placed in series with the compressor-turbine combination, as in a simple power turbine arrangement, the operating zone of the

compressor is limited still further.

The compressor and turbine characteristics charts are useful tools in analyzing and calculating the operating zones of the gas turbine engine. The compressor characteristics for the single compressor unit can be represented by three graphs as shown in Figure 4-1. The dashed line in each graph is the compressor surge line, and above this line, operation is unstable and impossible. The three graphs above can be collapsed into one useful picture (Figure 4-2) in which all the lines are related by the basic isentropic equations considering either isentropic or polytropic efficiency.

$$\frac{P_{02}}{P_{01}} = \left(1 + \eta_{cis} \frac{\Delta T_{02-01}}{T_{01}}\right)^{\gamma/\gamma-1} \tag{4.1}$$

$$\frac{P_{02}}{P_{01}} = \left(1 + \frac{\Delta T_{02-01}}{T_{01}}\right)^{\eta_{cp}\gamma/\gamma-1} \tag{4.2}$$

Turbine characteristics are, by far, simpler because their operation is more stable. The compressor moves fluid against the pressure gradient; and unstable operation, above the surge line, is where the flow stops because the angle of incidence becomes too great and separation occurs on the back of the blade. Consequently, the flow begins to reverse

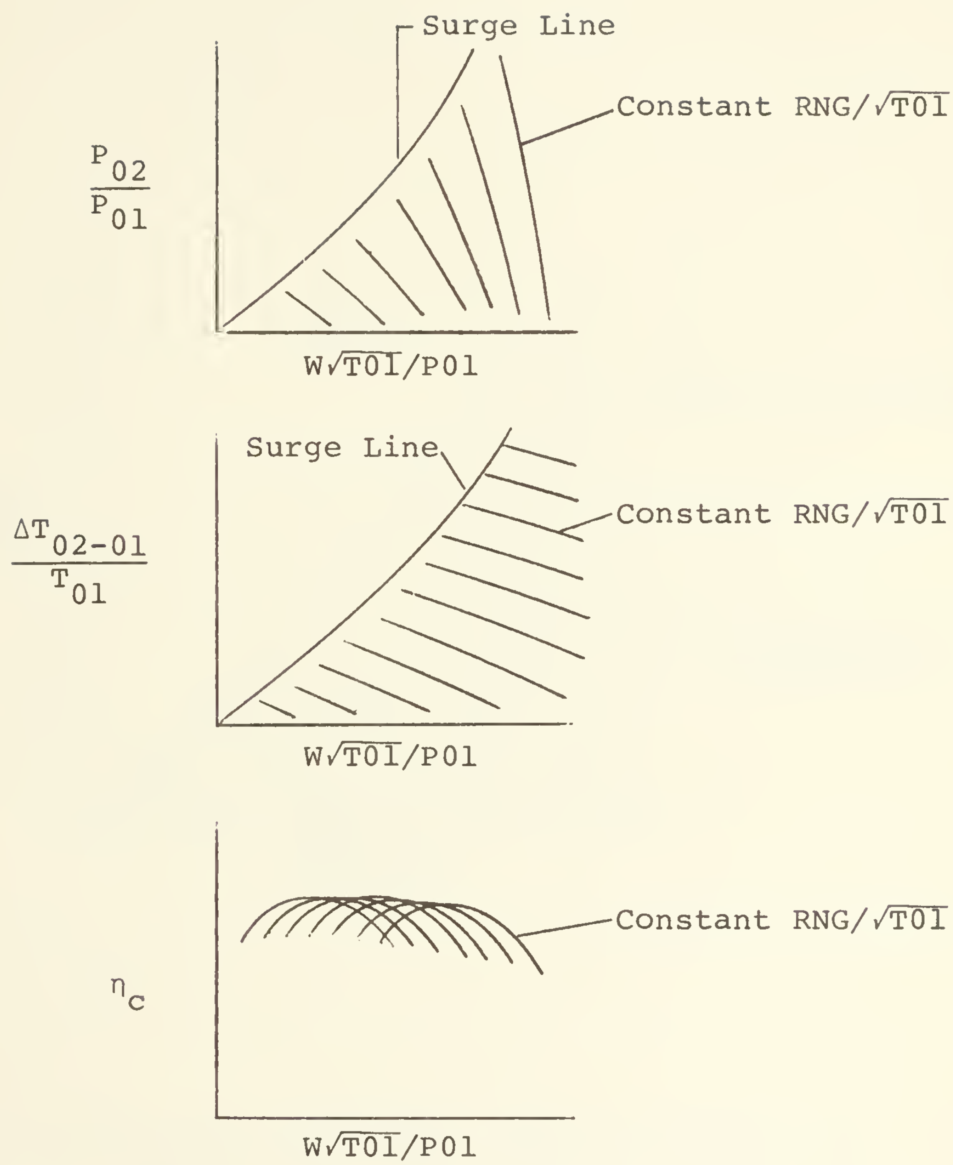


FIGURE 4-1: Standard Compressor Characteristics [4]

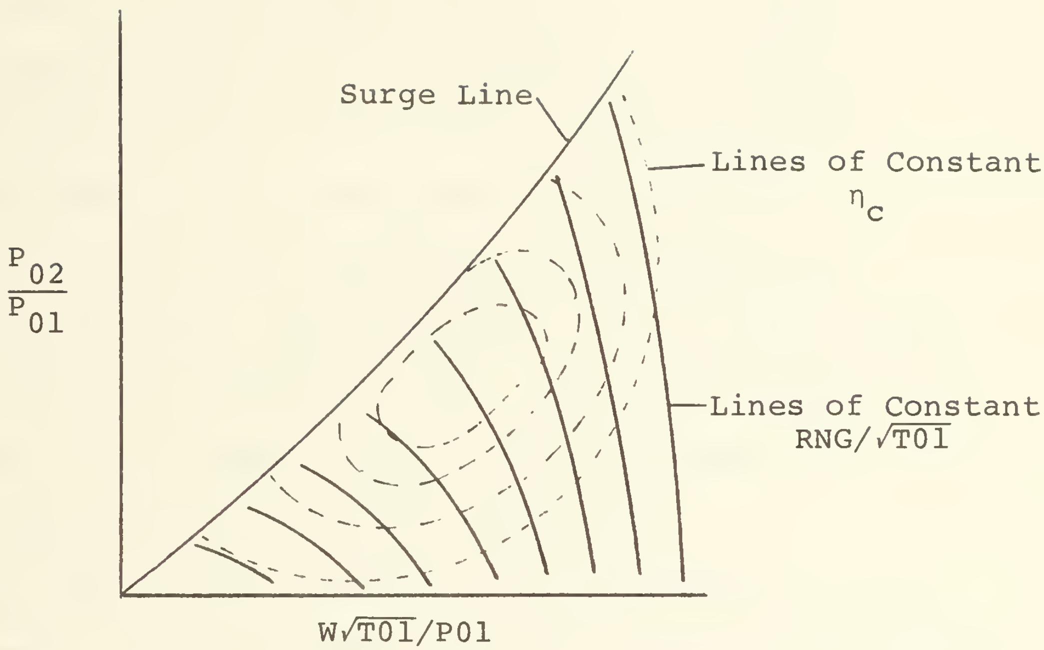


FIGURE 4-2: Complete Compressor Characteristics [4]

and go into a rotating stall, total stall, or surge depending upon the conditions. However, this paper is not concerned with unstable operation but with the stable, equilibrium operation beneath the surge line. A turbine, on the other hand, has a favorable pressure gradient in that the flow is in the direction from high to low pressure. Therefore, a turbine will always produce work. Turbine characteristics are shown below in Figure 4-3. However, normally, the variation of the efficiency of a turbine at any given pressure ratio is not large, particularly when the limited operation of a gas generator turbine is considered. So the resulting change in the temperature ratio is also not large, and therefore, the non-dimensional mass flow will vary insignificantly. The turbine characteristics, then, can be simplified as shown in Figure 4-4. This is a single line turbine characteristic.

Another way to express turbine characteristics is the "ellipse law". The ellipse law expresses the non-dimensional mass flow as a function of pressure ratio and assumes the efficiency and reaction do not change significantly off design.

$$\frac{W\sqrt{T_{03}}}{P_{03}K} = \sqrt{\left[1 - \left(\frac{P_{04}}{P_{03}}\right)^2\right]} \qquad \text{(see [4])} \qquad (4.3)$$

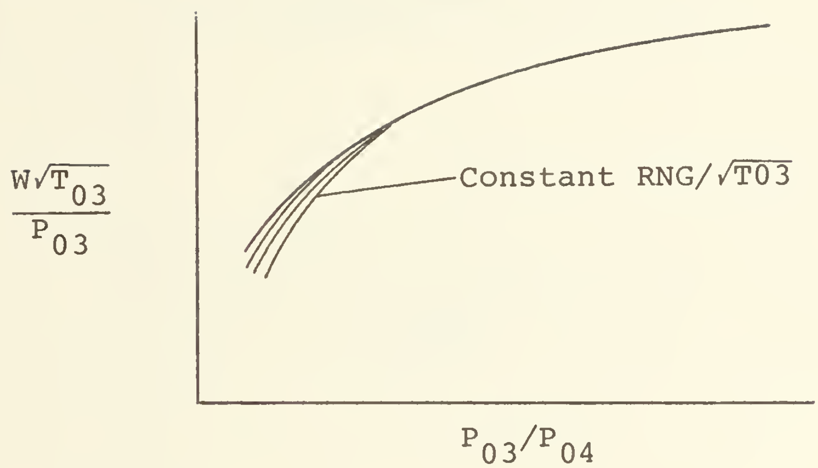


FIGURE 4-3: Standard Turbine Characteristics [4]

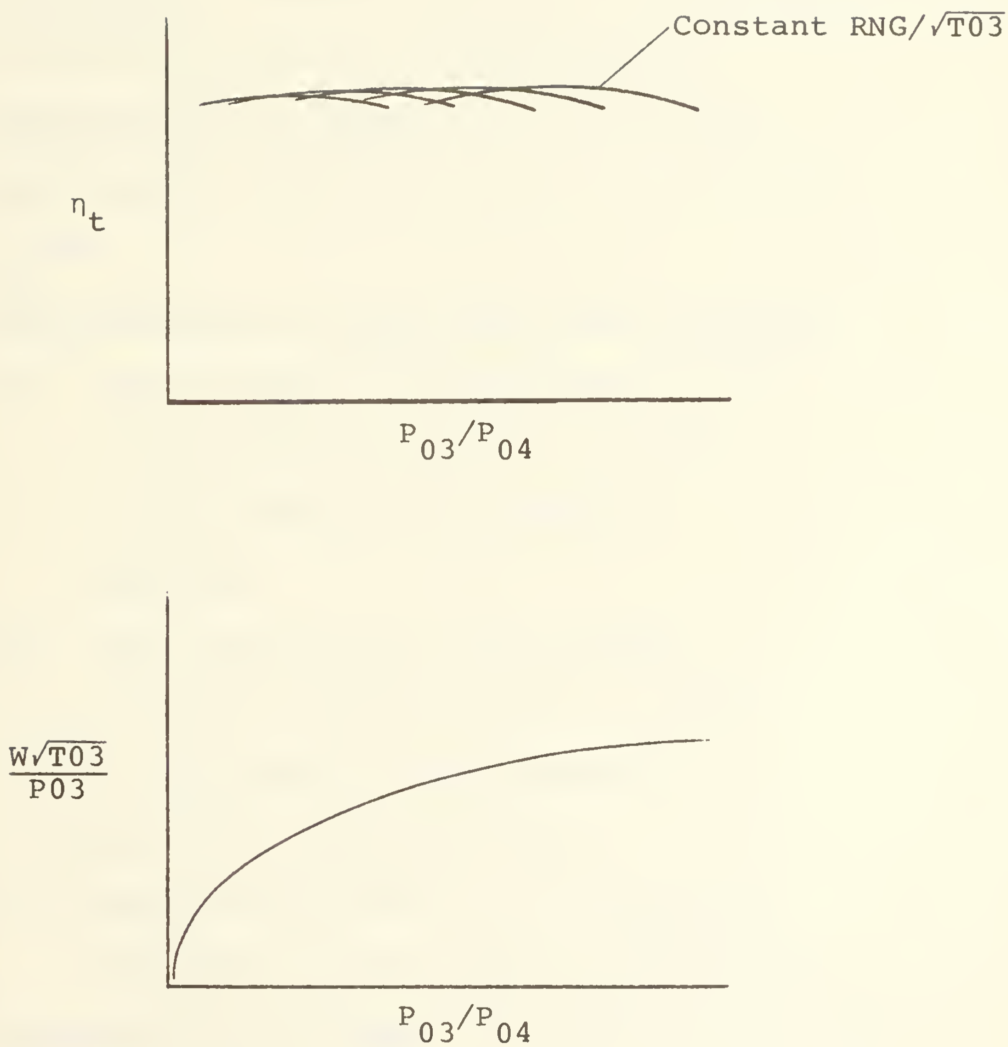


FIGURE 4-4: Single Line Turbine Characteristics [4]

The equation above is good for multistage turbines, and the "K" is a constant dependent on size of the machine. For single and two stage turbines, Table 4-1 has tabulated values of the mass flow for varying pressure ratios. It is a tabulation of the corresponding curves in Figure 4-5.

In calculating off design performance parameters, four basic rules are adhered to.

(1) <u>Continuity</u> - The mass flow into the component is equal to that out of the component.

(2) <u>Compatibility of Speeds</u> - Any components mechanically connected have equal rotational speeds.

(3) <u>Compatibility of Temperatures and Pressures</u> -

   (a) The pressure and temperature out of the downstream component is equal to that entering the upstream component.

   (b) The product of all the pressure increasing pressure ratios is equal to the product of the pressure decreasing pressure ratios.

   (c) Temperatures must vary consistently with work or heat input or extraction.

(4) <u>Compatibility of Powers</u> - In steady state, the

TABLE 4-1

[4]

$\frac{W\sqrt{T_{in}}}{P_{in}K}$	P_{in}/P_{out}		
	1 Stage	2 Stage	Multistage
0	1.0	1.0	1.0
0.1			1.0050
0.2	1.006	1.012	1.0206
0.3	1.020	1.033	1.0482
0.4	1.041	1.060	1.0916
0.5	1.070	1.097	1.1547
0.55	1.090	1.119	1.1999
0.6	1.113	1.149	1.2500
0.65	1.143	1.185	1.3159
0.7	1.178	1.227	1.4002
0.75	1.217	1.282	1.5119
0.8	1.270	1.356	1.6668
0.85	1.342	1.457	1.898
0.9	1.446	1.600	2.294
0.95	1.603	1.826	3.203
0.975	1.725	2.045	4.501
1.0	2 approx.	4 approx.	∞

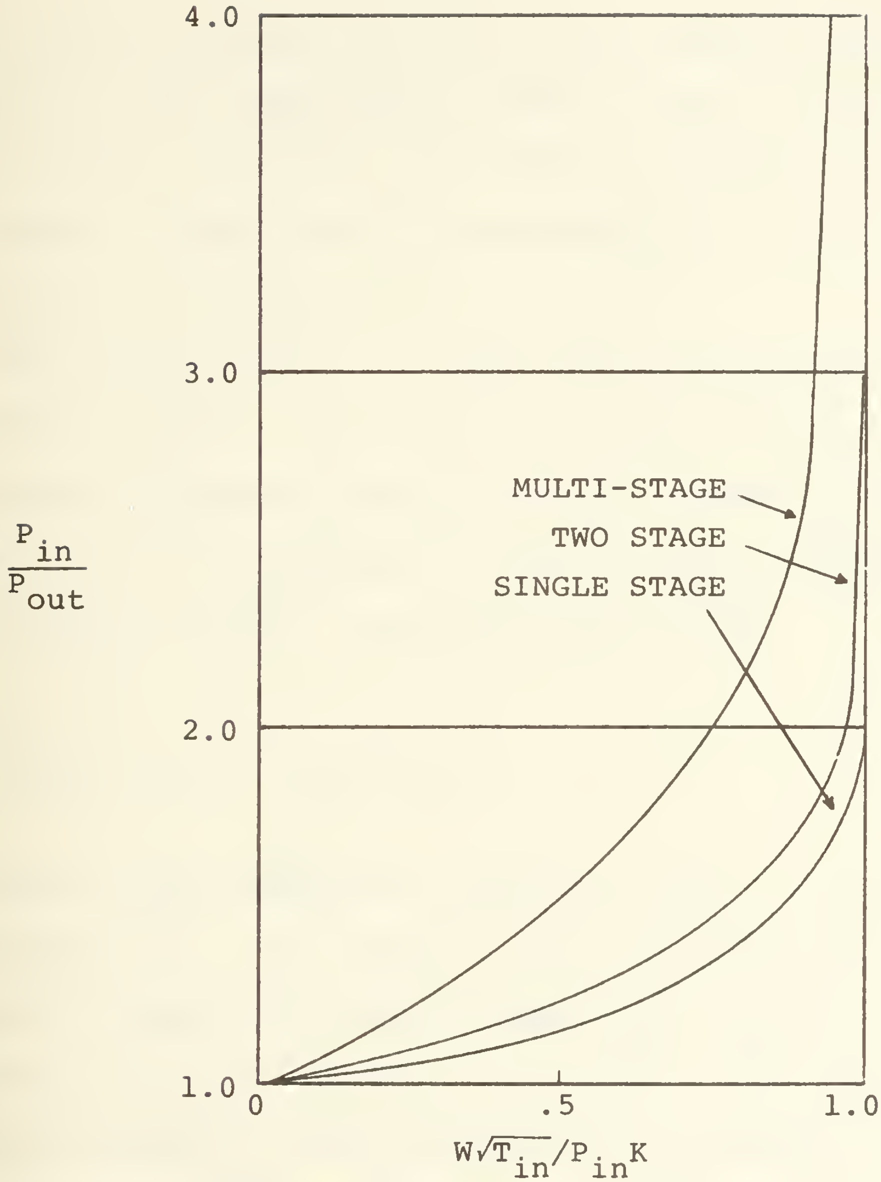


FIGURE 4-5: Graphical Ellipse Law [4]

powers produced and absorbed on the shaft are equal; ignoring losses.

In the actual calculations, it is also assumed that the mass flow in the compressor is equal to that in the turbines and the effects of losses and diffusers between components are ignored. These are valid assumptions as the data analyzation will bear out.

The gas generator in a free turbine engine is nothing more than a simple, open cycle gas turbine engine. The difference is that the work out of the turbine is equal to the work into the compressor. This can be expressed by the following relation (neglecting fuel flow).

$$\eta_t C_{p_g} \Delta T_{03-04} = C_{p_a} \Delta T_{02-01} \qquad (4.4)$$

By utilizing the compressor characteristics, it is possible to calculate T_{03}. If a point on the characteristics is arbitrarily selected, values of $RNG/\sqrt{T_{01}}$, P_{02}/P_{01}, $m\sqrt{T_{01}}/P_{03}$, and η_c can also be picked off. Then, using the isentropic relation with the addition of enthalpic efficiency, the temperature increase across the compressor can be calculated.

$$\Delta T_{02-01} = \frac{T_{01}}{\eta_c}\left[\left(\frac{P_{02}}{P_{01}}\right)^{\gamma-1/\gamma} - 1\right] \tag{4.5}$$

The difference between the gas generator calculations versus the simple, open cycle is that the pressure out of the high pressure turbine is not known. Consequently, it must be guessed. If P_{03}/P_{04} is then guessed, the value of $W\sqrt{T_{03}}/P_{03}$ can be obtained from the turbine characteristics. Now, making use of continuity of mass, T_{03}/T_{01} can be calculated.

$$\frac{W\sqrt{T_{03}}}{P_{03}} = \frac{W\sqrt{T_{01}}}{P_{01}} \times \frac{P_{01}}{P_{02}} \times \frac{P_{02}}{P_{03}} \times \sqrt{\frac{T_{03}}{T_{01}}} \tag{4.6}$$

And then using compatibility of speeds, $N/\sqrt{T_{03}}$ can be calculated.

$$\frac{RNG}{\sqrt{T_{03}}} = \frac{RNG}{\sqrt{T_{01}}} \times \sqrt{\frac{T_{01}}{T_{03}}} \tag{4.7}$$

With both $RNG/\sqrt{T_{03}}$ and P_{03}/P_{04} known, the efficiency of the turbine can be found by the turbine characteristic. Then $\Delta T_{03-04}/T_{03}$ can be calculated utilizing the isentropic relation for expansion in a turbine.

$$\frac{\Delta T_{03-04}}{T_{03}} = \eta_t\left[1 - \left(\frac{1}{P_{03}/P_{04}}\right)^{\gamma-1/\gamma}\right] \tag{4.8}$$

And this non-dimensional temperature drop can be used to calculate a new value of T_{03}/T_{01}.

$$\frac{\Delta T_{04}}{T_{03}} = \frac{\Delta T_{02-01}}{T_{01}} \times \frac{T_{01}}{T_{03}} \times \frac{C_{pa}}{C_{p_g} \eta_t} \qquad (4.9)$$

Equation (4.9) comes from rearranging Equation (4.3). The new value of T_{03}/T_{01} should agree with the previously calculated value, but if it doesn't, and it usually won't, a new value of P_{03}/P_{04} will have to be guessed. The iteration procedure should continue then until the two values of T_{03}/T_{01} agree within a tolerance. If this procedure is repeated numerous times, it would be possible to construct lines of constant T_{03}/T_{01} on the compressor characteristic chart. It follows that the compressor turbine provides just enough power to drive the compressor, so that there is one maximum cycle temperature for a given condition. These would specify the compressor operating range when coupled to that particular high pressure turbine.

In the final steps of calculation of off design parameters, the flow compatibility equation is utilized. The mass flow out the high pressure turbine is equal to that entering the power turbine.

$$\frac{W\sqrt{T_{04}}}{P_{04}} = \frac{W\sqrt{T_{03}}}{P_{03}} \times \frac{P_{03}}{P_{04}} \times \sqrt{\frac{T_{04}}{T_{03}}} \qquad (4.10)$$

If an operating point on the previously established lines is chosen from the compressor characteristics, the values of the right hand side of Equation (4.10) are known, and the non-dimensional mass flow to the power turbine can be calculated. Then with the power turbine characteristics, the actual value of the non-dimensional mass flow can be found from the pressure ratio, P_{04}/P_{05}, which is known. (P_{04} is known from the previous calculations and P_{05} is assumed atmospheric. Here, inlet and exit losses are ignored, but that is to keep the calculations clearer. Later, they will not be ignored.) If the two mass flow numbers do not coincide, then the compressor operating point chosen would not be an equilibrium point and the engine would accelerate or decelerate as necessary. If this is the case, new points must then be chosen from the compressor chart until the calculated non-dimensional mass flow equals that of the power turbine from the characteristic chart. Resulting from these laborious calculations is that for each non-dimensional speed of the compressor, there is only one point on that line at which the free turbine in operation with the gas generator will operate in equilibrium. Because of this fact, the operational zone of the compressor on its characteristics chart becomes one line; the equilibrium running line, shown in Figure 4-6.

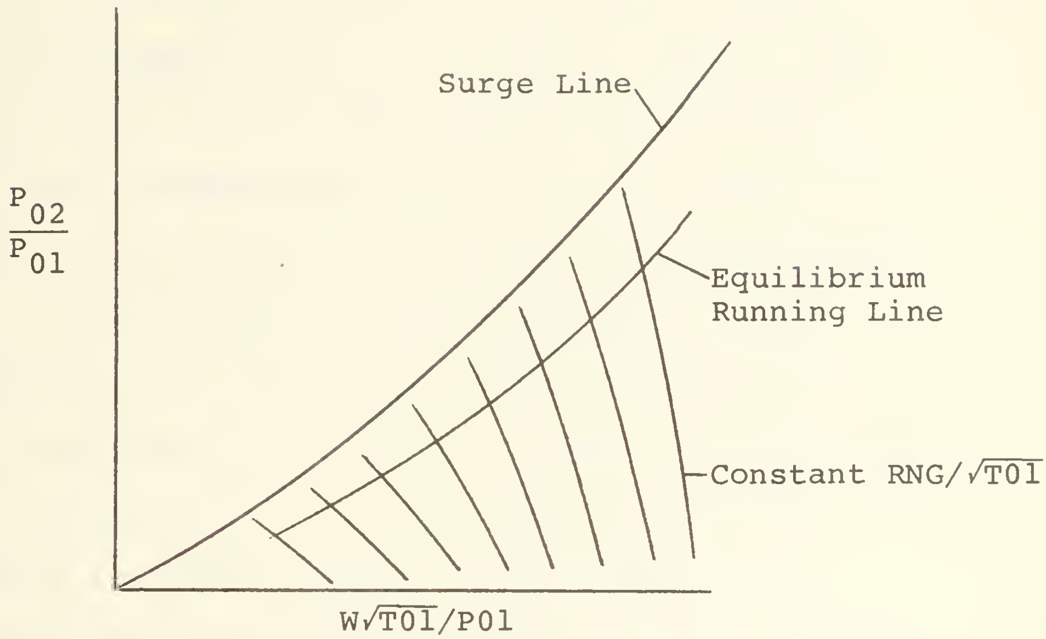


FIGURE 4-6: Free Power Turbine Equilibrium Running Line [4]

The iterative procedure mentioned above can be simplified considerably if the behavior of two turbines in series is considered. The effect of operating two turbines in this manner is shown in Figure 4-7. The curves show the characteristics of each turbine. In both graphs, the vertical scales, pressure ratio, are different, but the horizontal scales, mass flow, are identical. In the upper graph, the solid line is the input to the high pressure turbine, and the dashed line is its output, which is the input to the second, disregarding losses in the diffuser. The dashed line is established without the influence of the power turbine, but when the power turbine characteristics are plotted, as they are in the graph beneath the gas generator turbine, the two curves can be connected by a horizontal line based on continuity of mass. On both graphs, the dotted connecting line will determine equilibrium pressure ratios for both turbines. This condition is illustrated by the dotted lines connecting points (a) and (b) on the pressure ratio scales. But at point (a), the behavior is altered. This point is the choking point for the power turbine. The choking condition occurs when the velocity at the nozzle throat approaches the local sonic velocity. At this point, the non-dimensional mass flow remains constant while the pressure ratio increases. This

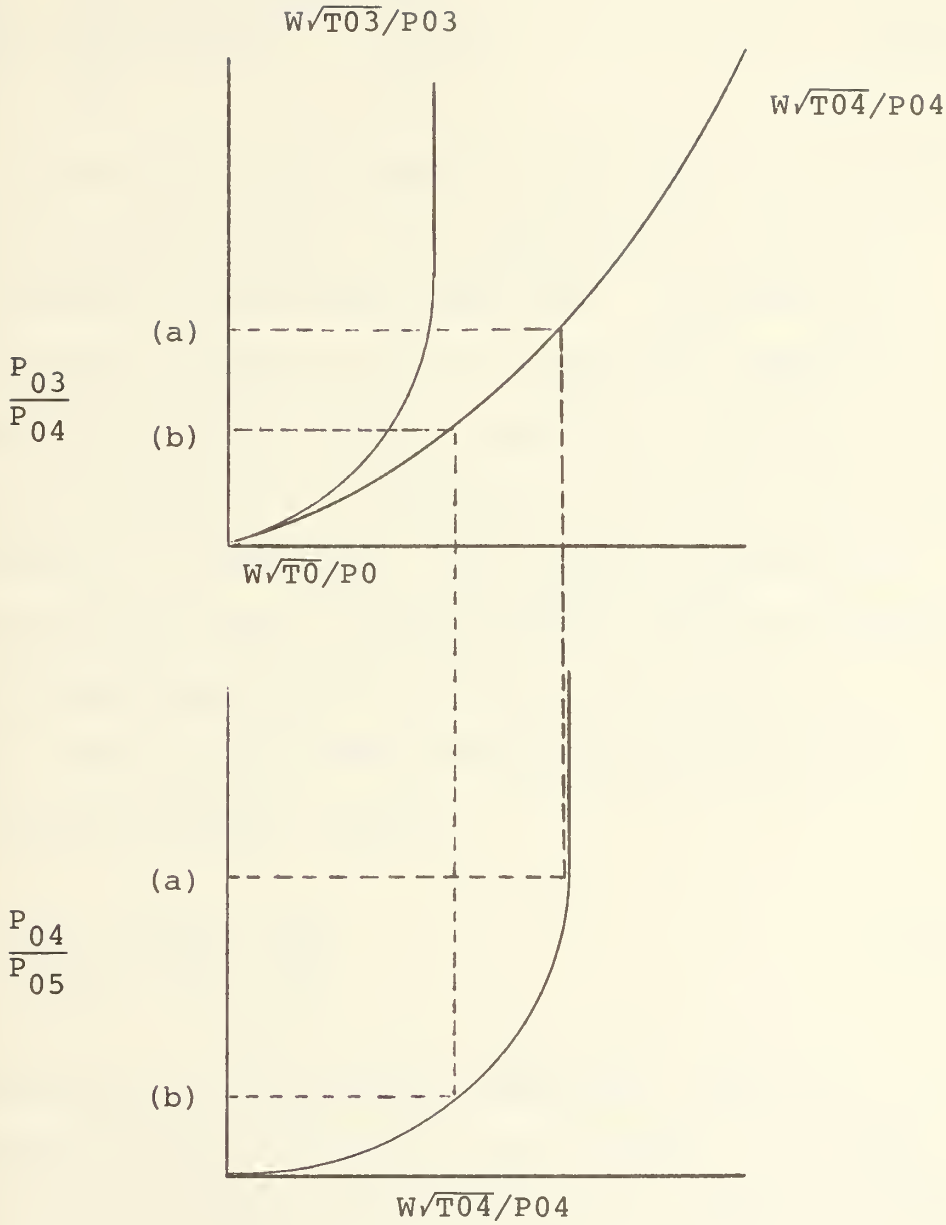


FIGURE 4-7: Free Turbine Coupling Effect [4]

condition is represented by the curves on the turbine characteristics. It is also represented by Equation (4.11) below (see [4]),

$$\frac{W\sqrt{T_{04}}}{A_4 P_{04}} = \sqrt{\left[\frac{\gamma g}{R}\left(\frac{2}{\gamma+1}\right)^{\gamma+1/\gamma-1}\right]} = K \qquad (4.11)$$

where A_4 is the total area normal to the flow at the nozzle throat of the power turbine. The expression on the left remains constant in the choked condition. Therefore, as long as the power turbine is run in the choked condition, the pressure ratio across the gas generator turbine is fixed, and if the design is economical, it will be near its choked condition. Graphically, this can be expressed as a constant pressure ratio across the high pressure turbine for varying compressor pressure ratios, as shown in Figure 4-8. This can be established by considering the compatibility of pressure ratios.

$$\frac{P_{03}}{P_{04}} = \frac{P_{03}}{P_{02}} \times \frac{P_{02}}{P_{01}} \times \frac{P_{05}}{P_{04}} \qquad (4.12)$$

As the pressure ratio across the compressor increases, so do the pressure ratios across the two turbines. But once the power turbine becomes choked and P_{03}/P_{04} becomes constant, the continuing pressure increase in the compressor is

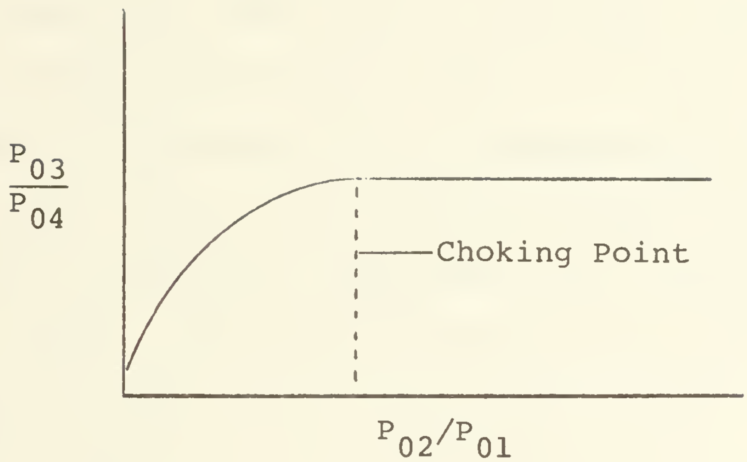


FIGURE 4-8: Free Turbine Pressure Ratio Relationship [4]

countered by an equal exit pressure increase in the power turbine, considering the combustion process at constant pressure. In other words, and most importantly, the pressure ratio of the compressor; and therefore its rotational speed because it is linear with pressure ratio; is independent of the output of the free turbine running in a choked condition.

Another consequence of operating the power turbine in the choked condition is that T_{03}/T_{04} is a constant also. Since the non-dimensional mass flow of the power turbine is constant, the pressure ratio across the high pressure turbine is fixed and therefore, its non-dimensional mass flow is fixed also. Taking the ratio of the mass flows yields the result of constant temperature ratio across the gas generator turbine. The non-dimensional mass flows are also related by the area ratio if both turbines are operated choked. This can be shown by Equation (4.11).

$$\frac{W\sqrt{T_{04}}}{A_4 P_{04}} = \frac{W\sqrt{T_{03}}}{A_3 P_{03}} \tag{4.13}$$

Since by continuity the mass flows are equal, Equation (4.13) becomes

$$\frac{\sqrt{T_{04}}}{\sqrt{T_{03}}} \times \frac{P_{03}}{P_{04}} = \frac{A_4}{A_3} \tag{4.14}$$

and temperature and pressure ratios for a particular turbine are constant.

The choked condition has been mentioned throughout the preceding discussion. A power turbine and gas generator turbine are designed to operate in a choked condition for as much of the power range as possible. The reason for this is that operation is more efficient. The gas turbine, then, is more efficient at high speeds, which is well known and can be shown easily. Looking back at Figure 4-6, if the operation is dropped off from the choked condition, for example from (a) to (b), there is a larger percentage of pressure ratio change in the power turbine. Consequently, more of the available pressure is absorbed in the high pressure turbine relative to the power turbine. From the isentropic expansion relation,

$$\left[\frac{P_{02}}{P_{01}}\right]^{\gamma-1/\gamma} = \left[\frac{T_{02}}{T_{01}}\right] \qquad (4.15)$$

if more pressure is absorbed, then there is a greater change of temperature across the turbine; and from the compatibility of temperatures, the work must be correspondingly larger. So, as the operation comes down from the choked condition, more of the available work is absorbed by the gas generator on a percentage basis. This means the efficiency is

decreased because the summation of the work is less for a constant heat input. This is shown by the equation for the thermal efficiency,

$$\eta_t = \frac{\Sigma W}{Q_A} \tag{4.16}$$

where work and heat are per pound mass of fuel. The choke point corresponds to the value of $m\sqrt{T_{04}}/P_{04}K$ equal to 1.0 using the ellipse law. From Figure 4-5, it is possible to increase the PR across the turbine while the mass flow remains constant. In this case, the flow becomes supersonic, and no additional work is achieved. As the pressure ratio is increased further, the efficiency goes to zero. However, this is not possible in the power turbine engine because once the power turbine becomes choked, the pressure ratio across the high pressure turbine is set. By compatibility of pressures, the pressure ratios across the compressor and power turbine are also set. So the power turbine operates at $m\sqrt{T_{04}}/P_{04}K$ equal to 1.0.

In the discussion of the operation of the free power turbine engine, several key factors have been established and will be used to establish the equilibrium running line for the LM2500 and in the discussion on monitoring. The first half of the chapter stated that when a compressor and

turbine are operating as a gas generator, the compressor is restricted to a family of operating lines on the compressor characteristics chart. Then when a power turbine is added, the compressor is restricted to a single operating line, the free turbine equilibrium running line. The second half of the chapter went into a more detailed study of the consequences of running two turbines in series. The first of these was that the non-dimensional mass flows at the throats of the high pressure and power turbine nozzles are related by the area ratio when both are operating in a choked condition. Secondly, the ratios P_{03}/P_{04} and T_{03}/T_{04} are constant when the power turbine is choked. And this means that the output of the power turbine is independent of compressor speed and pressure ratio. Finally, the most efficient operation of the engine is with both turbines operating in a choked condition.

CHAPTER V

MONITORING THEORY AND PROCESS

Monitoring an engine is a process which involves determining changes in vital parameters and comparing them to baseline data. Over a period of time, some parameters will increase their percent of deviation from the baseline, and when the deviation is plotted against time, a trend can be seen. Trends in certain sets of parameters indicate specific problems in various components of the engine. The key to an informative trending program is to monitor the parameters which are most affected by resulting problems and will point to the specific problem area. In addition, they should be able to point to problems in the control system or indicate whether or not the engine is properly tuned. To obtain changes in operation, one variable has to be chosen which will be the baseline abscissa parameter. This variable should be the major input to the control system.

There are two basic types of control systems for marine gas turbine engines. They are designated "speed-control" and "power-control". The "speed-control" system has as its primary sense the rotational speed of the power turbine. The basic schematic for the system is shown in Figure 5-1. In addition to sensing the power turbine speed, there is

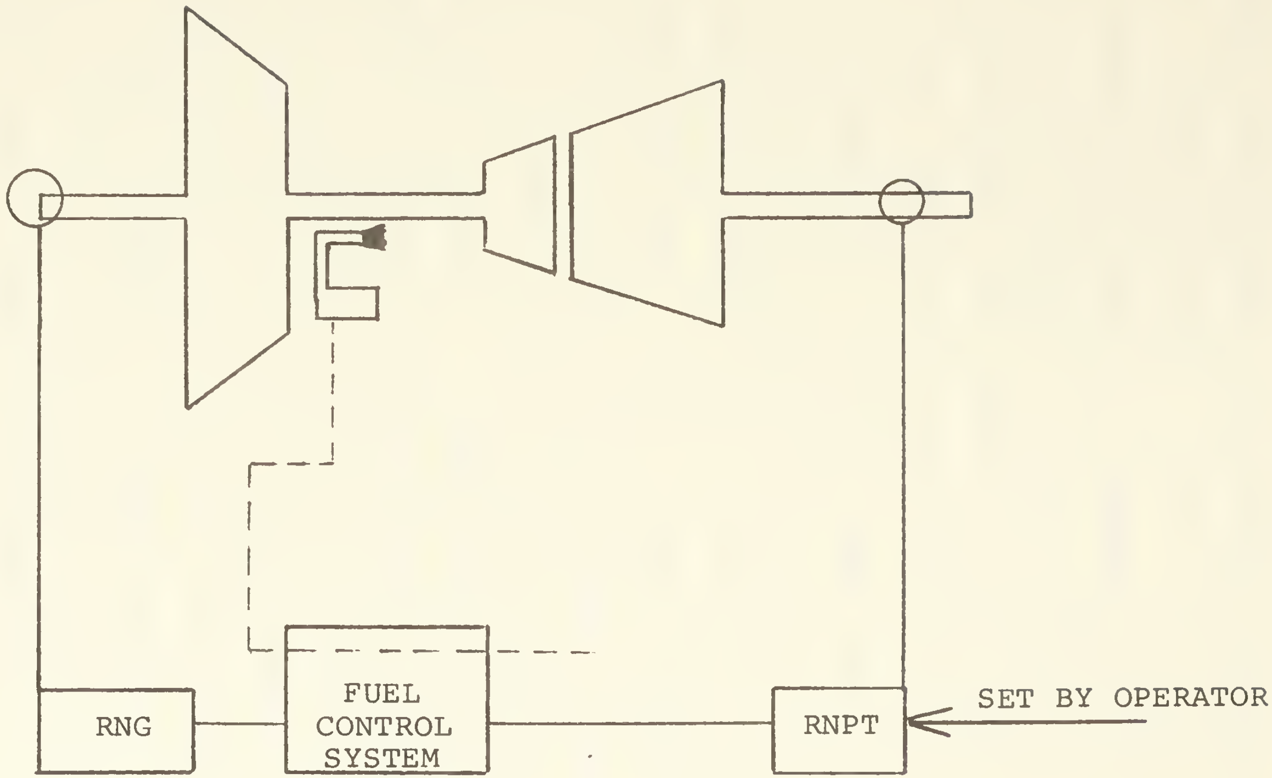


FIGURE 5-1: Speed Control Schematic

also a component which senses the rotational speed of the gas generator and has overriding control, but this is only to ensure that the gas generator is operating within limits. "Speed-control" is not common in control systems for propulsion gas turbines. The response of the system is more sensitive than that of the "power control", but it is also slower. Consequently, following seas; which cause the flow velocity through the propellor to fluctuate; and rough seas; which lift the propellor out of the water; are going to cause the gas generator to cycle. These conditions result in more wear on systems and an increase in fuel consumption. An engine in this situation would rarely, if ever, attain steady operation. "Speed-control" does have a use in control of electric power generators. "Power control" does not exhibit the problems mentioned above, but instead, in adverse situations, the gas generator will run at a constant speed while the power turbine speed fluctuates. A basic schematic of "power-control" is shown in Figure 5-2. Using the "power-control" system allows the free turbine to find its equilibrium speed which is essential for the monitoring procedure in this discussion. This system is the basis for the control of the engines in the Spruance class destroyer above twelve knots. Since the speed of the gas generator is the primary controlling factor,

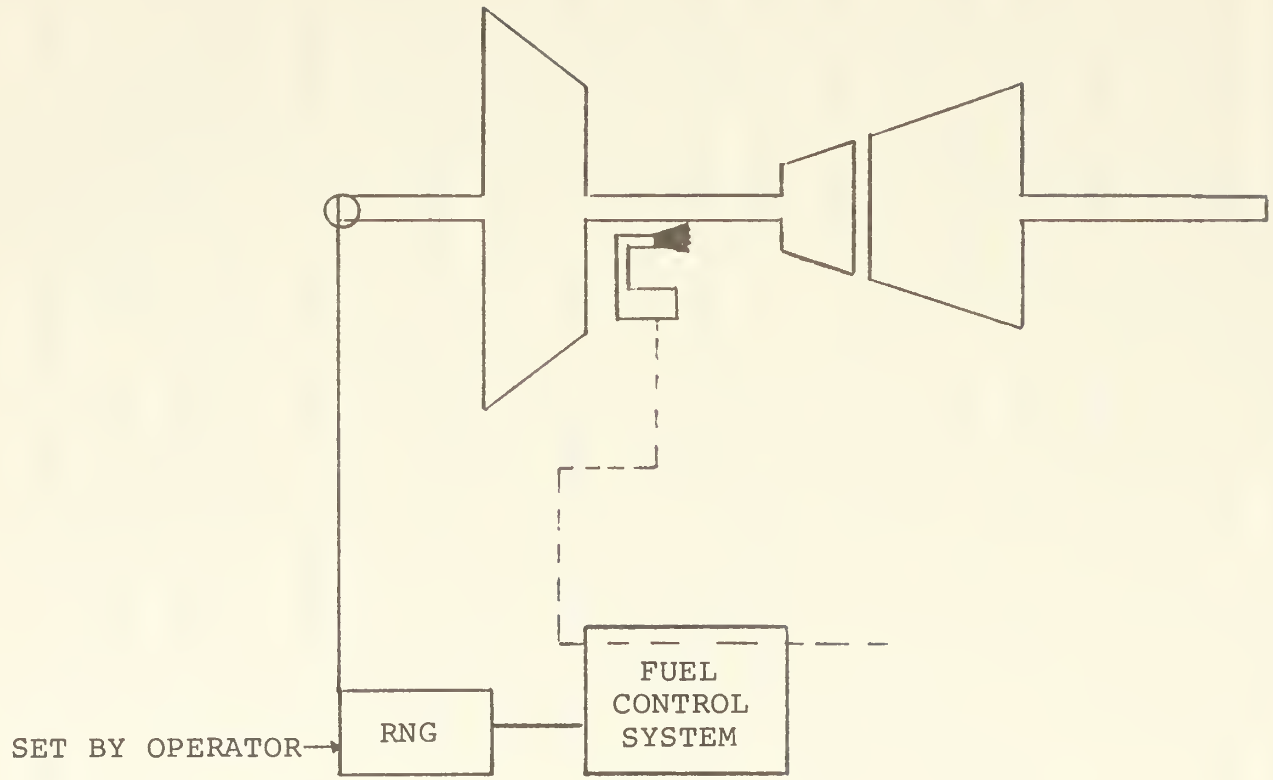


FIGURE 5-2: Power Control Schematic

it will be the baseline abscissa parameter, and the evaluation procedure then is carried out for a value of gas generator RPM.

The control system does not specifically dictate the choice of gas generator RPM for the baseline abscissa. It could have been pressure ratio across the compressor or across the gas generator, but if this was the case, the evaluation process would be different. With choice of RPM, the evaluation thought process parallels that of the fuel control system. The system is told to attain an RPM, and if the high pressure turbine performance is degraded, the control system will have to put in more fuel to obtain the desired speed. In addition, it was established in Chapter IV that the gas generator RPM is independent of the power output of the engine. Since the power output of a marine gas turbine is nearly completely controlled by fuel flow, which is controlled by the RPM, the speed of the gas generator is the independent variable in the system.

The power level of the engine or gas generator speed setting is attained by a lever angle actuated by an operator. Between zero and twelve knots the speed of the ship is controlled by the pitch of the propellor while the output shaft runs at constant speed. Over the speed of twelve knots, the pitch is held constant, and the shaft speed is

varied. These relationships are illustrated in Figure 5-3. The trending procedure discussed in this chapter is useful only in the range over twelve knots. Below this range, the power turbine is no longer choked, and the engine is in a maneuvering mode utilizing speed control where efficiency and economy are of no consideration. In Chapter VI, it will be established that the procedure is useful in the horsepower range from 10000 to 25000. This range is sufficient for a trending device where long term deteriations can be detected. However, for on condition monitoring, it would be necessary to cover the entire horsepower range.

The preceeding discussion, concerning control, is not meant to be comprehensive. There are other inputs to the control of fuel flow besides gas generator speed, but considering the variations for a measured RPM allows qualitative changes in the parameters to be determined.

The determination of parameters to monitor involves a consideration of what deficiencies can occur. It is practically impossible to consider all deficiencies, but a few of the more common problems can be reviewed. The next step is to evaluate whether or not the parameters will correlate to the operating efficiencies of all the engine components. And this is probably the most significant,

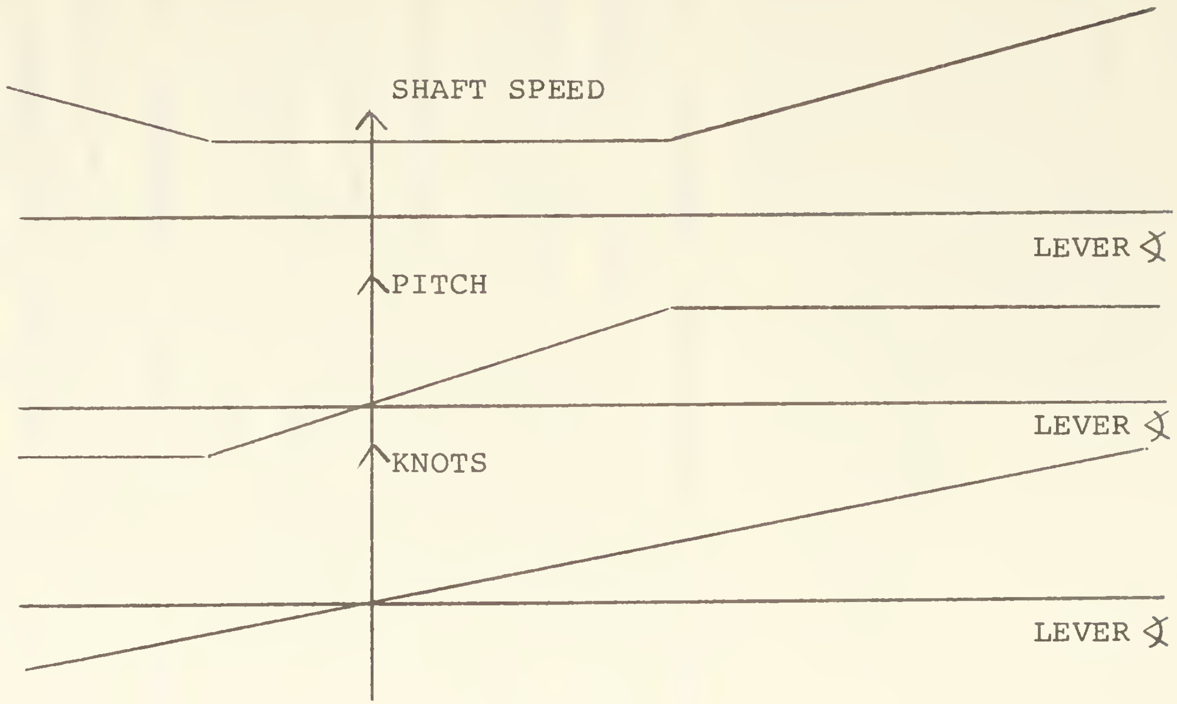


FIGURE 5-3: Power Control Relationships

because it is this fact that allows the interpreter to determine where the problem lies. The parameters chosen are listed below.

Compressor pressure ratio. . . . . PRC

Specific fuel consumption. . . . . SFC

HP turbine pressure ratio. . . . . HPTPR

Power turbine inlet temperature. T054

Pressure ratio across the gas generator. PRGG

Brake horsepower . . . . BHP

A common deficiency for a marine gas turbine is a fouled compressor. In this case, the control system is going to specify an RPM for the gas generator, and in the compressor's degraded condition, it will be unable to produce the pressure ratio that correlates to the RPM attained. The decrease in pressure ratio will also cause a decrease in compressor outlet temperature, and the fuel flow will decrease because it will take less work to attain the RPM desired. Considering compatibility of pressure ratios, the gas generator outlet pressure must decrease since the compressor discharge has decreased. The result is a lower pressure ratio across the gas generator and a lower pressure to the power turbine. Consequently, the pressure ratio of the power turbine is decreased, dictating a decrease in the

high pressure turbine pressure ratio, which will increase the gas generator outlet temperature. The power turbine expands to essentially atmospheric pressure, so the decrease in inlet pressure means a decrease in temperature ratio and therefore a decreased horsepower production. The qualitative changes in monitored parameters are summarized in Table 5-1. The degradation of the compressor causes its outlet conditions to change which results in decreasing pressure ratios and work across the turbines. The first indication of fouling is the decrease in PRC and PRGG, and allowing the problem to progress will result in a decrease in horsepower for the speed of the gas generator. The change in SFC will not be significant. One assumption that is made throughout the evaluations, and which was stated in Chapter IV, is that the efficiency of the turbines remains constant over the varying conditions when not degraded. This is a valid assumption and will be shown to be the case in Chapter VI. The efficiency affects the temperature ratio of the turbine, and if it remains constant, the decrease in pressure ratio determines a corresponding decrease in temperature ratio.

$$\frac{P_{02}}{P_{01}} = \left(1 + \frac{1}{\eta_t} \frac{\Delta T_{021}}{T_{01}}\right)^{\gamma/\gamma-1} \qquad (5.1)$$

Equation (5.1) illustrates the trend, and the idea is used in all monitoring evaluations.

Problems in the high pressure turbine produce different variations in parameters. When turbine fouling occurs, the pressure ratio across the compressor will be normal, therefore producing an unchanged air flow. The control system will request an RPM, but because of the drop in efficiency of the turbine, more fuel will be required to obtain the RPM. This will increase the maximum cycle temperature, and the drop in turbine efficiency will further increase the gas generator outlet temperature. The pressure ratio across the gas generator turbine will decrease causing the total gas generator pressure to decrease. Since the turbine fouling has altered the turbine characteristics, the power turbine will remain choked with no change in temperature or pressure ratios. Therefore, the horsepower production will not change. The outlet conditions of the power turbine will increase, however. The increased fuel will cause an increase in SFC. Worn seals will indicate similar parameter changes as a fouled turbine. The fouling results in a higher enthalpy outlet which is analogous to worn seals, where there is an enthalpy loss through the seals. The losses between the stages will maintain no change in pressure ratio, but the decrease in mass flow will

result in a decrease in horsepower. Bowed nozzle vanes in the HP turbine will print out a signature identical to the fouled HP turbine. In monitoring the turbine, the initial indication of a problem is the increasing gas generator outlet temperature, indicating that the turbine is removing less energy from the fluid. The result is to increase the fuel to increase the energy per pound of fluid. As the problem persists, the pressure ratio across the turbine will decrease, but the horsepower production will not be altered. The power turbine inlet is high in pressure and temperature, but the mass flow has not been altered because the compressor is not degraded. The change in pressure ratio across the HP turbine does not influence the power turbine as it would if the turbine were not degraded. Gas generator degradation signatures are shown in Table 5-1 for comparison with compressor degradation.

Monitoring of the combustor is reflected by the change of SFC. The combustor has virtually no moving parts and relies on the air flow and the nozzles to atomize the fuel so that proper combustion is achieved. The most effective method of monitoring is to record the temperature distribution from several temperature probes. A variation in temperature around the annulus then points toward a faulty combustion unit or can. However, as all of the nozzles wear

evenly over a period of time, the distribution of the temperature will be steady, but the SFC of the engine will increase due to incomplete combustion as a result of poor atomization. More fuel will have to be added to achieve the normal maximum cycle temperature and power to drive the compressor. The condition does lead to quicker turbine fouling, which will then show up as previously discussed. But the other engine parameters will remain unchanged because the fuel flow is increased to attain the normal operating points of the other components of the system. Combustor degradation can be compared to other components in Table 5-1.

Degradation of the power turbine will be reflected by a drop in horsepower for the gas generator RPM. The drop in horsepower will cause an increase in SFC. The operation of the gas generator will remain unaffected because operation of the power turbine only affects the distribution of pressure ratio across the high pressure turbine, and for small degradations, it can be assumed that the power turbine remains choked, resulting in constant HP turbine pressure. Table 5-1 illustrates a degraded power turbine signature.

In the foregoing discussion, two assumptions have been made to simplify the reasoning in constructing Table 5-1. The additional fuel addition has a negligible effect on the

TABLE 5-1

SIGNATURES OF COMPONENT DEGRADATION

PROBLEM	SIGNATURE						COMPONENT
	PRC	SFC	HPTPR	T054	PRGG	BHP	
Fouled Compressor	↓	↔	↓	↑	↓	↓	Compressor
Fouled HP Turbine	↔	↑	↓	↑	↓	↔	High Pressure Turbine
Worn HP Turbine Seal	↔	↑	↔	↑	↔	↓	High Pressure Turbine
Bowed HP Nozzle Vanes	↔	↑	↓	↑	↓	↔	High Pressure Turbine
Fouled Power Turbine	↔	↔	↔	↔	↔	↓	Power Turbine
Combustor Nozzles Worn	↔	↑	↔	↔	↔	↔	Combustor

↑ ≡ +

↓ ≡ -

↔ ≡ 0

mass flow, and this is valid since the mass of fuel is less than 5% for stoichiometric combustion. And secondly, the combustion process is at constant pressure. This is necessary because in actual practice the combustor exit temperature and pressure are not measurable.

It can be seen from Table 5-1 that each component will produce a distinct signature when problems develop. In actual practice, the procedure would be reversed. Here, the problems were considered and the changes evaluated. In monitoring the engine, the signatures are given and the problem has to be evaluated. Using the parameters selected, the reverse procedure is not difficult. The indication of correct operation of the compressor is the pressure ratio developed, PRC. Pressure ratio is a linear function of RPM, and with non-varying stator vanes, the function is one line. The primary indicator of high pressure turbine degradation is the gas generator outlet temperature. And this applies to the power turbine also; except that BHP is the monitored parameter. When its performance is degraded, the BHP will decrease. However, it can be seen that the BHP decreases for various problems in other components of the engine. Thus the evaluator must be able to understand basically the inter-relationships between the components if a correct diagnosis is to be made. For a degraded compressor,

the pressure ratios downstream are going to decrease, and this, coupled with a lower mass flow, is going to yield a lower horsepower for the gas generator RPM developed. When the high pressure turbine becomes degraded, the parameters of the components downstream from the compressor are going to change while the pressure ratio across the compressor is normal. A problem in the combustor will show up as an increase in SFC only while the remaining parameters remain unchanged. And finally, a power turbine drop in efficiency will have no effect on the other components and therefore, will only be reflected in the changes in SFC and BHP. Any component degradation will cause an increase in SFC, and this is a good indicator of the overall efficiency of the engine. A limiting change in SFC could be imposed to indicate or predict overhaul. In addition, it could be useful in prediction of engine life for extended deployments where support facilities are not available. The most important parameter to monitor is the pressure ratio across the high pressure turbine. It should remain constant and will indicate when the engine is operating at the most economical equilibrium point for the desired RPM. It is independent of load and is therefore an indication of maximum efficiency regardless of propellor pitch or damage. And it holds over any region of operation as long as the

power turbine remains choked.

The trending discussed so far has involved common deficiencies which affect the four major components of the engine. Other components include the control system and the monitoring equipment. In addition, the problems previously discussed show up as gradual changes in operation over a long period of time. Some problems, however, will plot as immediate changes in the trends. An example of this is a damaged gauge or one that has been replaced without calibration. When this occurs, only one parameter will change, and it will do so in a step fashion. Previously, at least two parameters changed (except in the combustion chamber problem). Other examples of step changes are brittle cracks appearing in bleed ducts or combustion chamber liners. Foreign object damage to one of the major rotating components would cause a step degradation and appear as a loss of component efficiency. Control system problems could occur as step trends if the system suddenly went out of calibration or a component failed. Or a gradual wearing of the linkage could produce a trend of increased fuel and therefore SFC. Control system problems would yield signatures similar to the engine signatures, and the problem then would have to be evaluated by other monitoring methods as discussed in Chapter II.

The basis of the parameter monitoring procedure is to record the changes of parameters periodically and plot them as a percent of deviation versus time. The accuracy of the measuring instruments is very important and is responsible for the trending. Degradation in the system will only result in parameter changes on the order of one to three percent deviation. One reading is not significant, but instead it will take many over a long period of time to indicate a trend. If the instruments are more precise, the trend can be obtained earlier and the curve will appear smoother. The shapes of the curves will then point to engine deficiencies.

CHAPTER VI

THE OPERATIONAL BASELINE

The baseline data are the actual operating points for the engine in steady state conditions. This information is programmed into the computer module and is considered normal operation. It is then from this data that the deviations from the normal are computed by comparing the actual engine parameters to the baseline for a particular speed of the gas generator. The data used to obtain the baseline is from Reference [1], which is a listing of engine parameters measured by varying the speed of the power turbine while the gas generator runs at equilibrium to produce a particular horsepower. The parameters were measured at different horsepower levels, and the complete runs were repeated at different inlet temperature settings. Data was generated between -65°F and 130°F and from horsepower settings of 5000 to maximum. Standard atmospheric conditions were 59°F and 14.696 PSIA. There was no altitude or humidity factor considered and no water flow to the engine. The data was collected on a test stand absorbing the generated power by a water brake. Consequently, there was negligible inlet and exit duct losses, and the brake has the effect of varying the pitch of the propellor. From the data, it appears that the RPM of the power turbine was the

controller set point. This fact has no bearing on the monitoring sequence discussed in Chapter V, where it was established that the logic of the engine control and monitoring is based on gas generator RPM control, because the system is in steady state and operating at a single point on the equilibrium running line. The torque generated by the shaft at the brake will be nearly equal to that generated by a controllable pitch propellor operating in open water, and therefore, the RPM of the test will correspond to the RPM of the shaft with a propellor at full pitch, ignoring gearing and shaft losses. The baseline was established at 59°F and 14.696 PSIA (International Standard Organization Day) over the horsepower range from 10000 to 25000, in increments of 5000. The data used is included in the appendix. The bulk of the operating points was found graphically and refined by utilizing the theory established in Chapter IV.

In the interest of economy, the steady state equilibrium operating points were based on minimum SFC. From the data, graphs were constructed for each horsepower rating of SFC versus RPM of the gas generator (RNG) as the independent variable. Then it was possible to pass a curve through the points, and the lowest point of the curve related the minimum SFC and the corresponding gas generator speed. For

most of these graphs, it was difficult to obtain the actual minimum SFC, but the RPM could be obtained within a half of a percent. These graphs are shown in Figures 6-1 through 6-4. Then, with the RPM for each horsepower specified, graphs of other parameters could be constructed. At 10,000 horsepower, graphs were drawn of T_{054}, P_{054}, WF (fuel flow), RNPT (power turbine RPM), and torque; all versus the independent variable RNG (Figures 6-5 through 6-9). Then, knowing the gas generator speed, the values of the dependent parameters could be picked off. The graphs of torque and NPT were nearly linear in the range of gas generator RPM, so for the remaining horsepower, linear interpolations would be adequate. Graphs were then constructed of the three parameters for each of the remaining horsepowers (Figures 6-10 through 6-18). In addition to torque and power turbine RPM, PS3, W2, and W8 were obtained by interpolation using the gas generator speed as the basis, and the first set of baseline data was known. The results are summarized in Table 6-1.

If the power turbine is choked, the pressure ratio across the high pressure turbine is constant. For now, the power turbine is assumed choked. Figure 6-19 is a graph of the high pressure turbine pressure ratio, HPTPR, for the operating range from Table 6-1 (assuming constant pressure

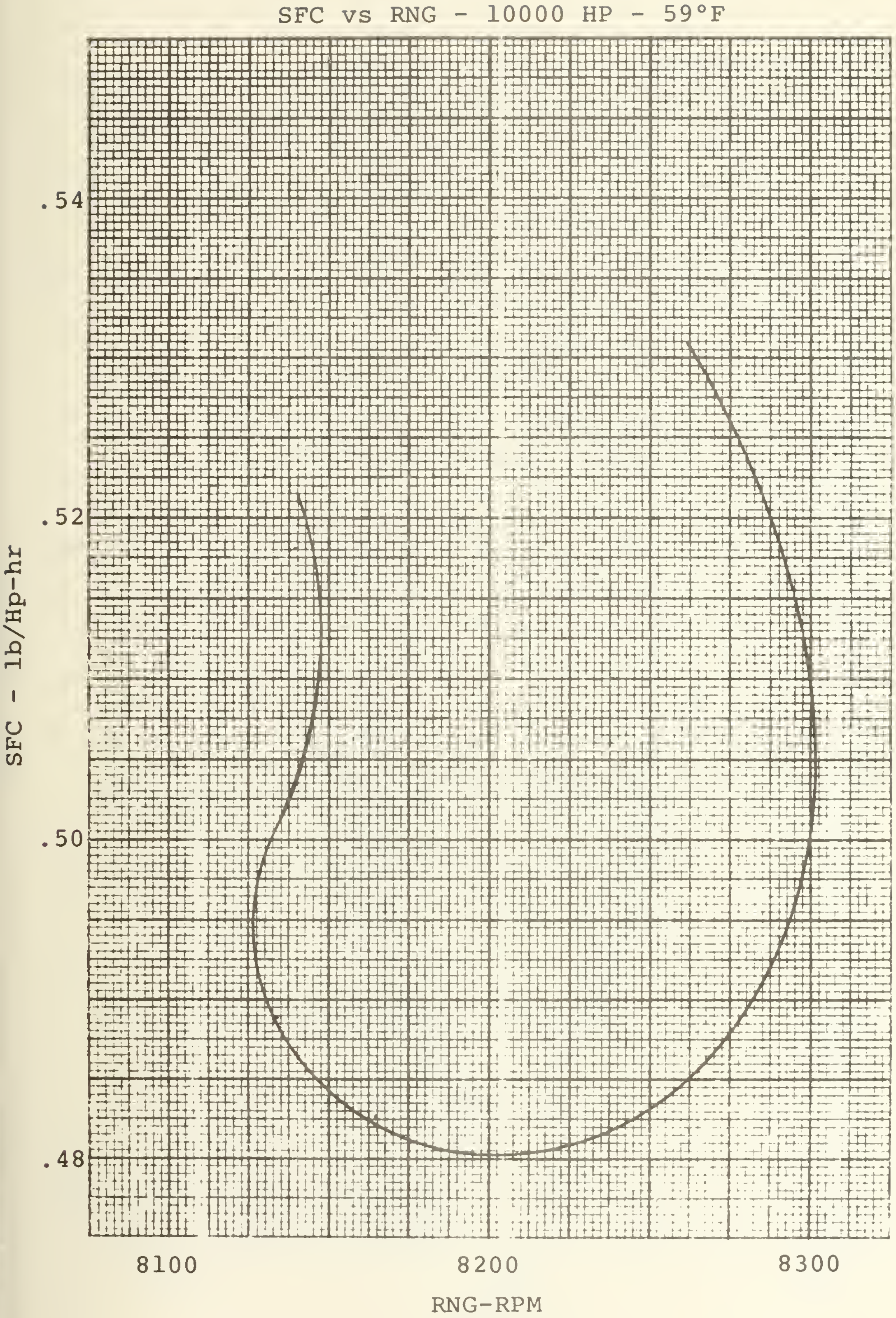

SFC vs RNG - 10000 HP - 59°F
.54
.52
.50
.48
SFC - lb/Hp-hr
8100
8200
8300
RNG-RPM

FIGURE 6-1

SFC vs RNG - 15000 HP - 59°F

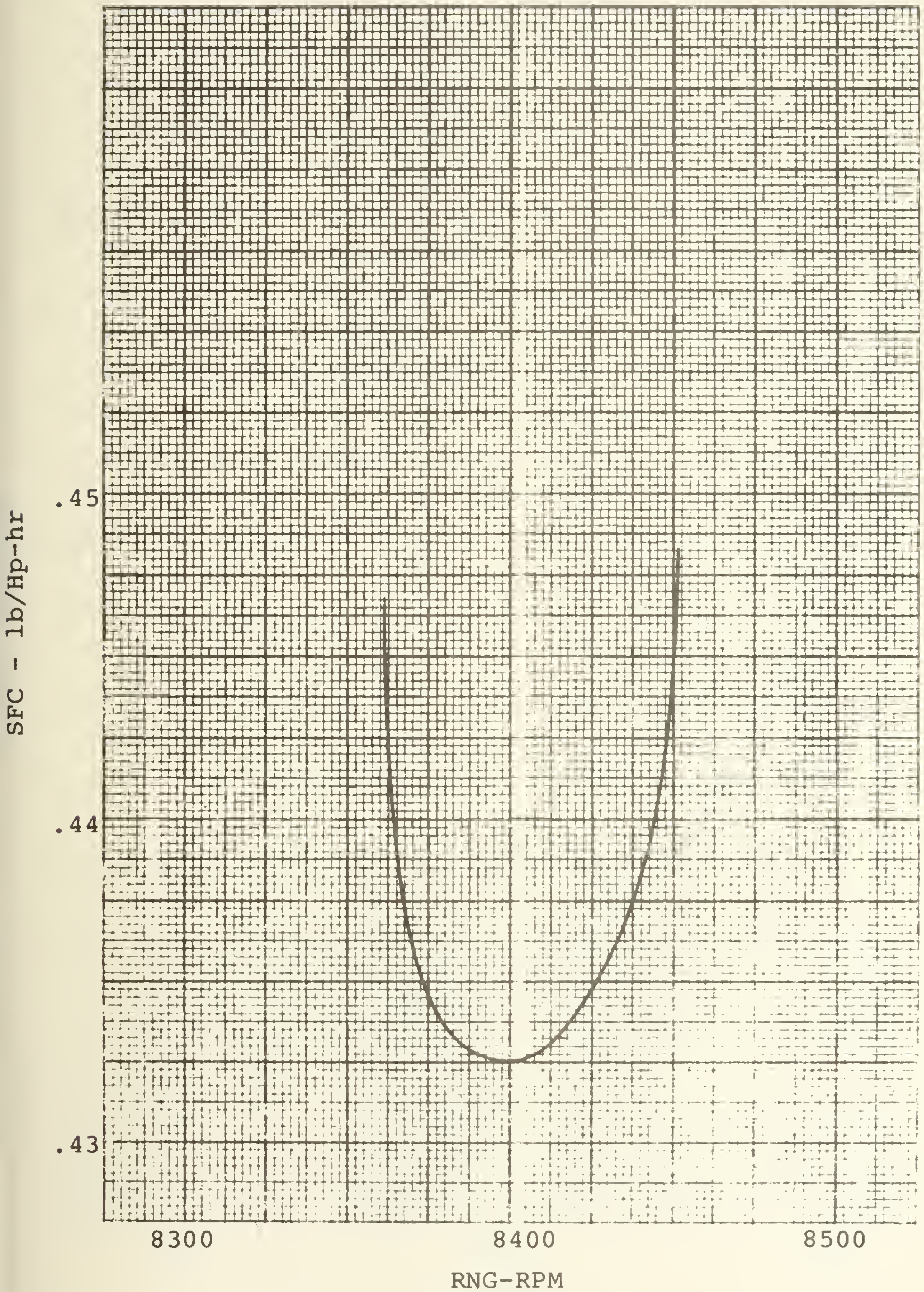


FIGURE 6-2

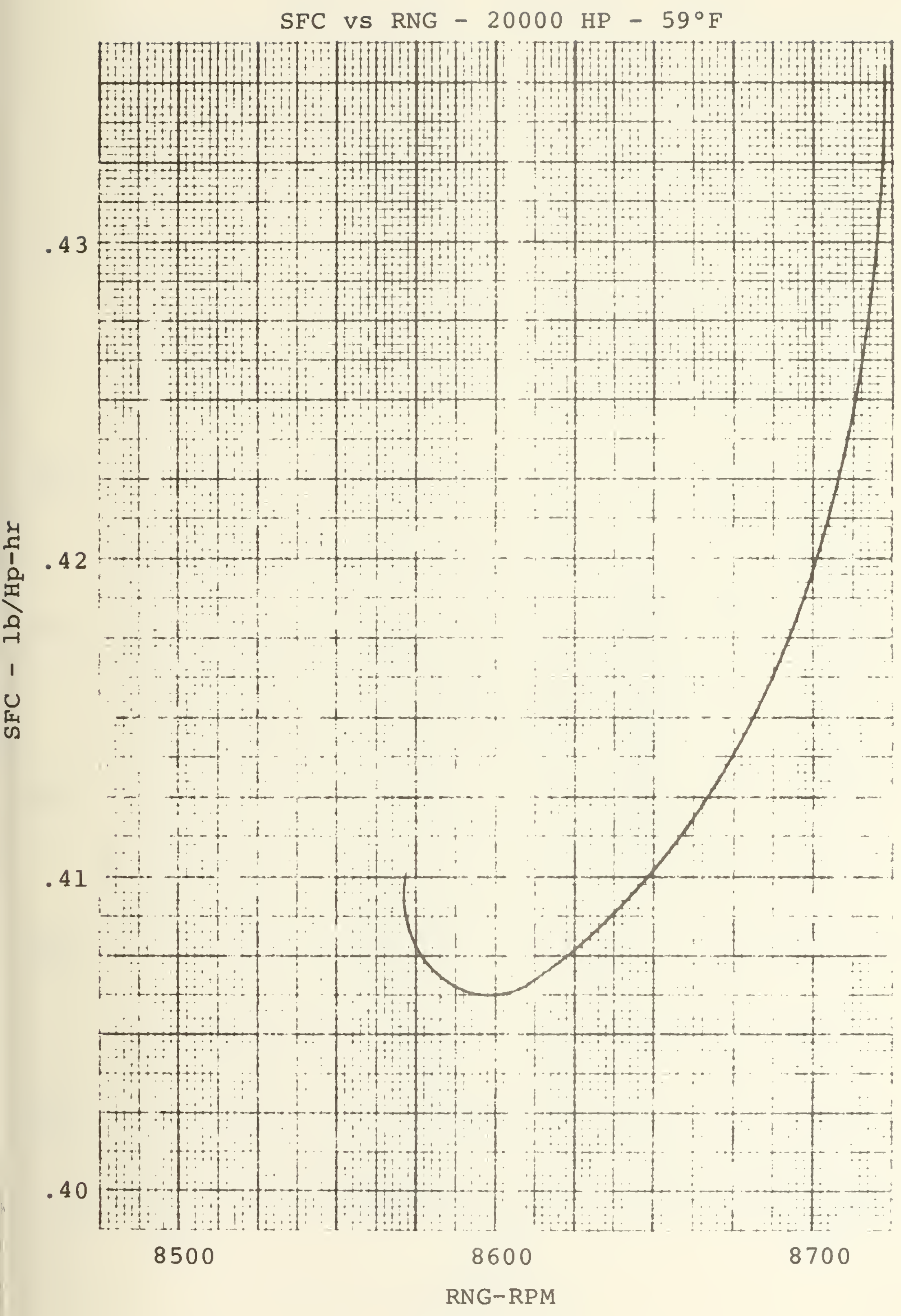

SFC vs RNG - 20000 HP - 59°F
SFC - lb/Hp-hr
.43
.42
.41
.40
8500
8600
8700
RNG-RPM

FIGURE 6-3

SFC vs RNG - 25000 HP - 59°F

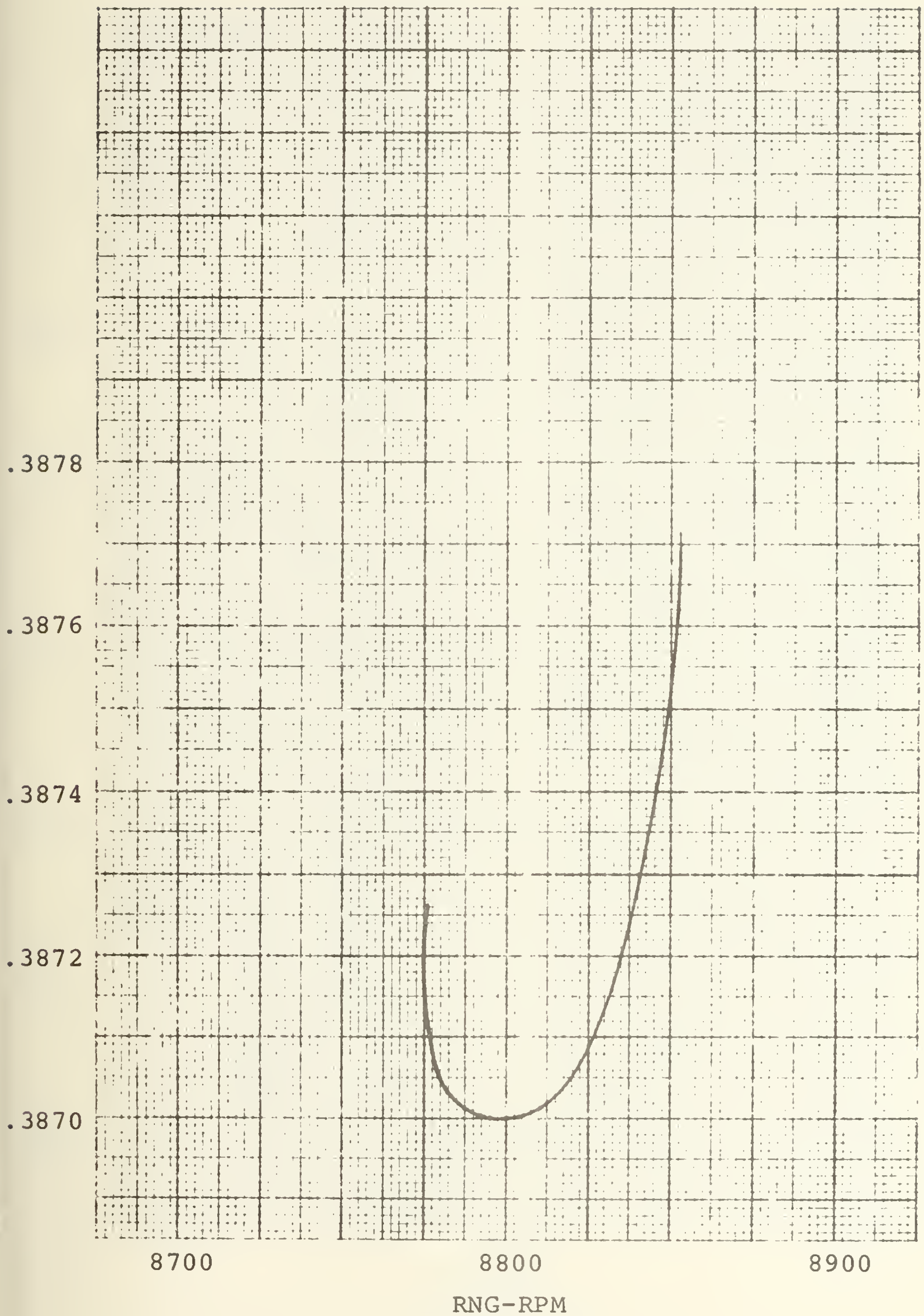


FIGURE 6-4

T054 vs RNG - 10000 HP - 59°F

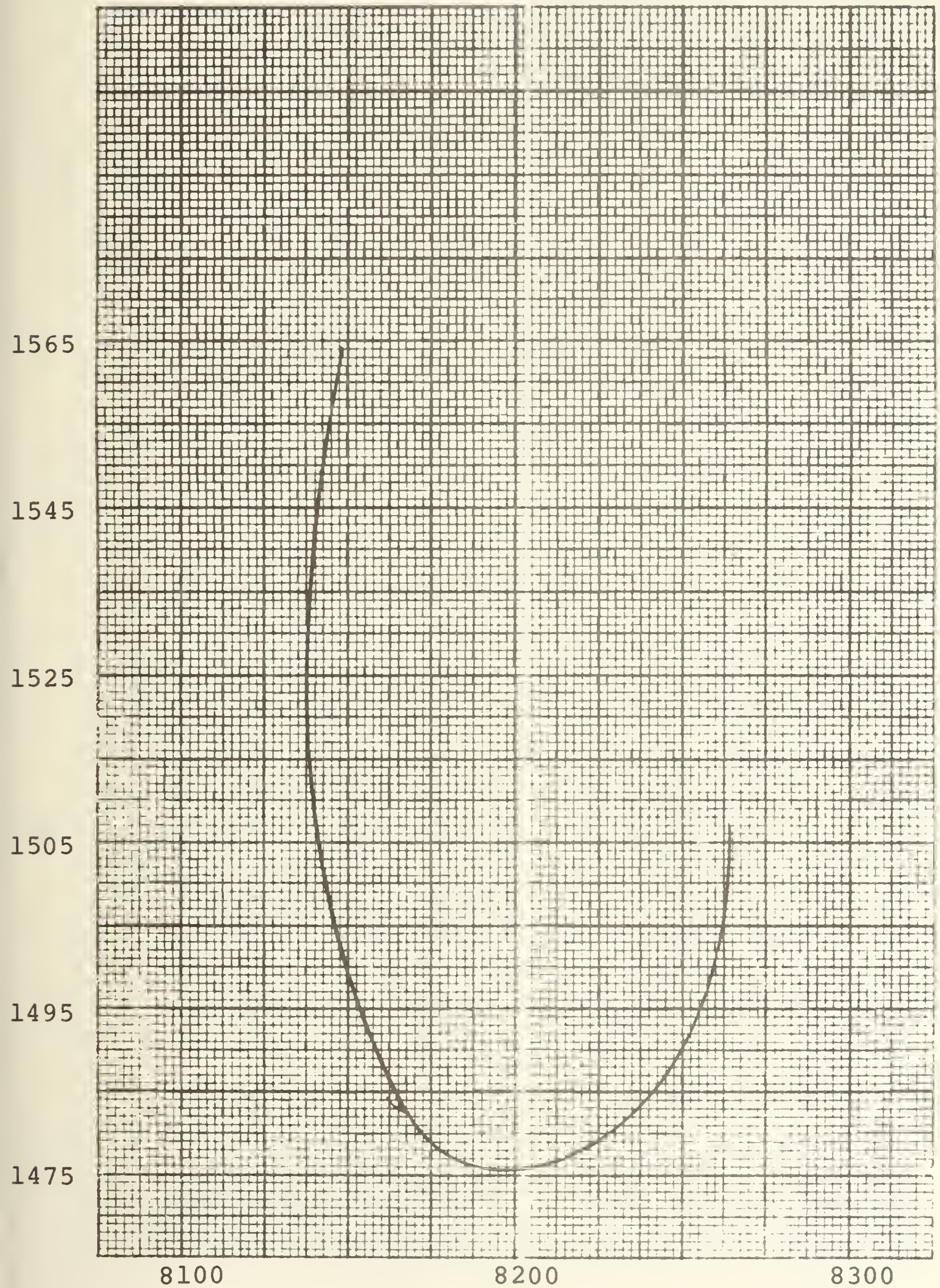


FIGURE 6-5

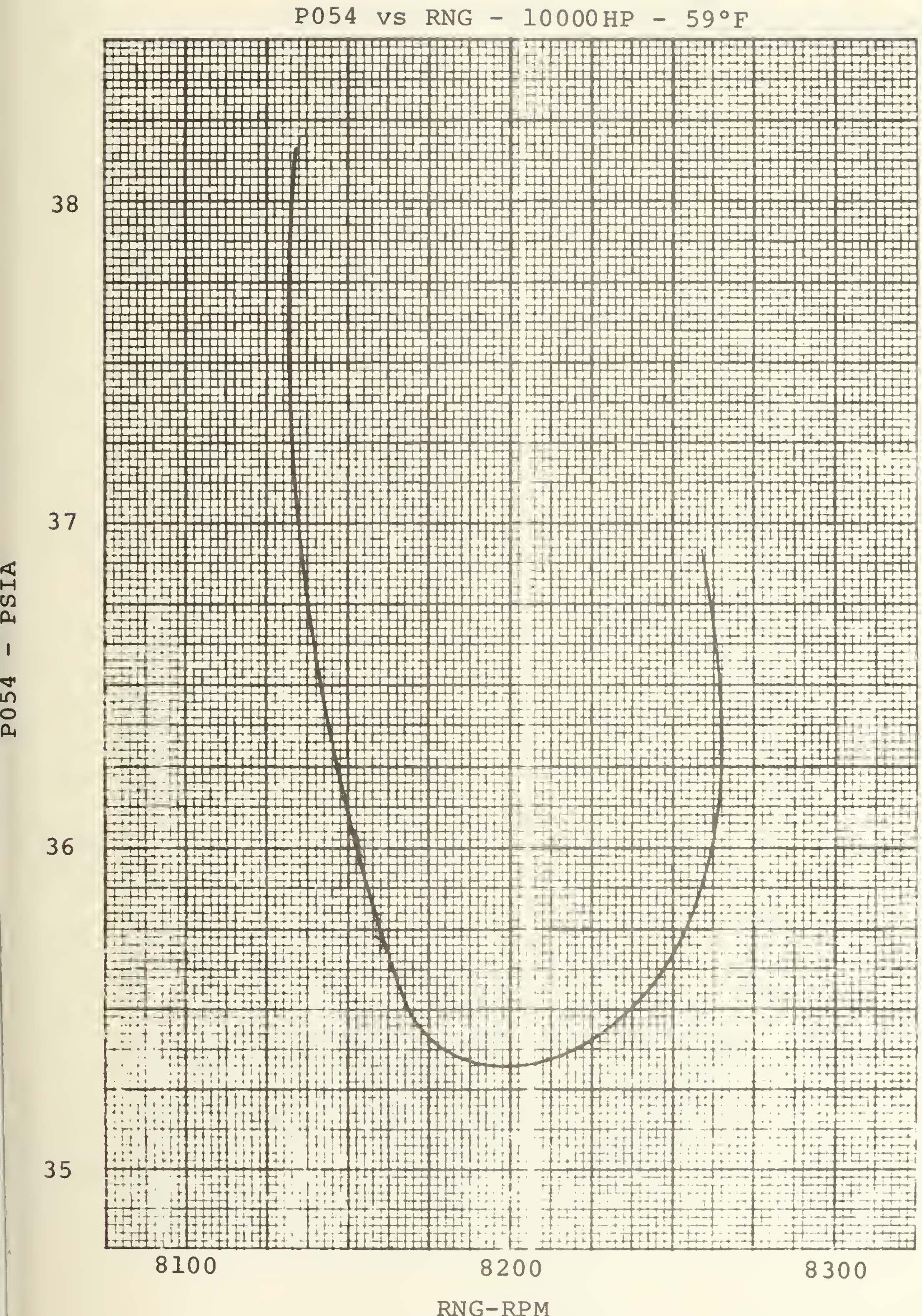

P054 vs RNG - 10000 HP - 59°F
38
37
36
35
P054 - PSIA
8100
8200
8300
RNG-RPM

FIGURE 6-6

WF vs RNG - 10000 HP - 59°F

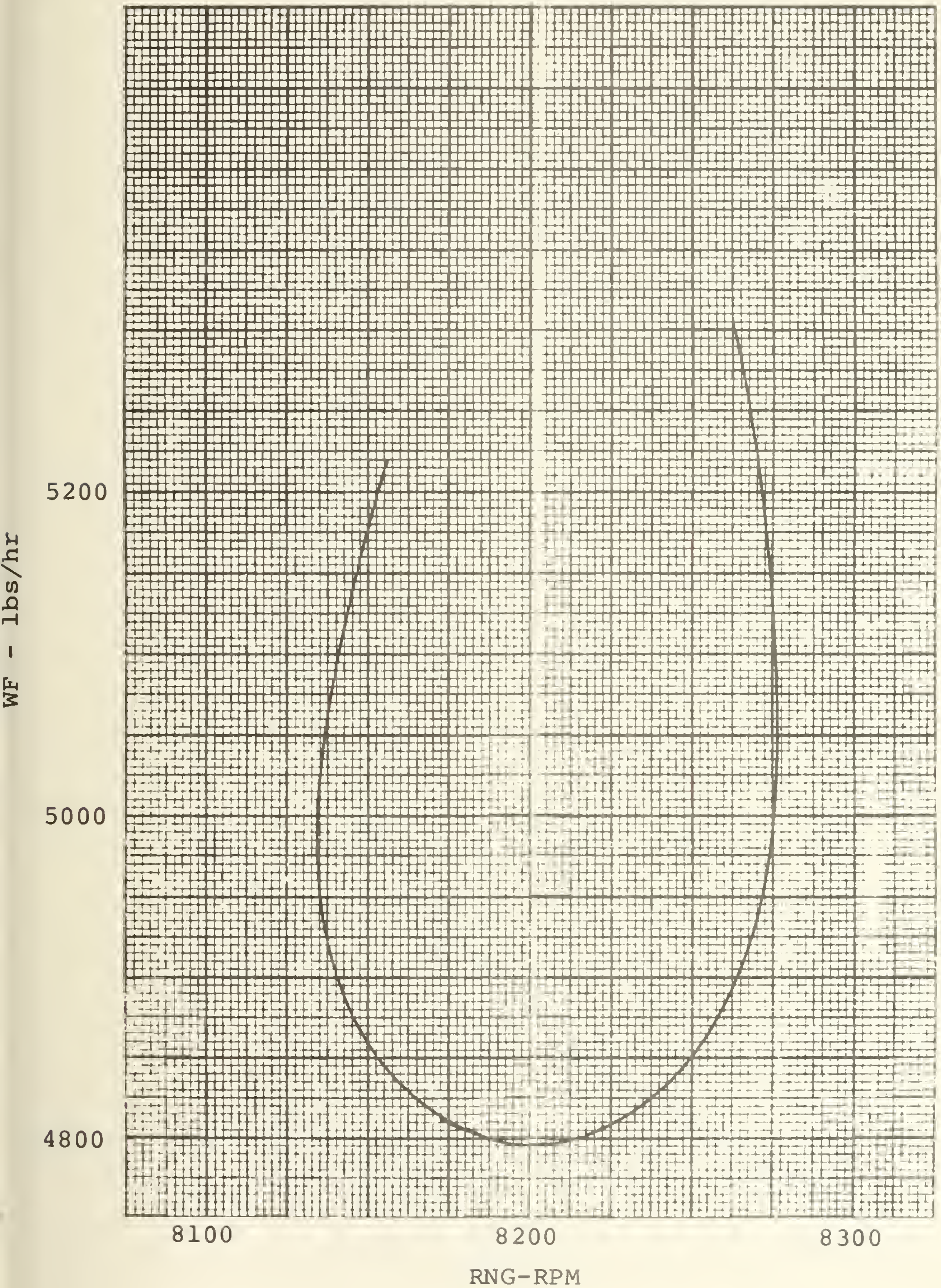


FIGURE 6-7

RNPT vs RNG - 10000 HP - 59°F

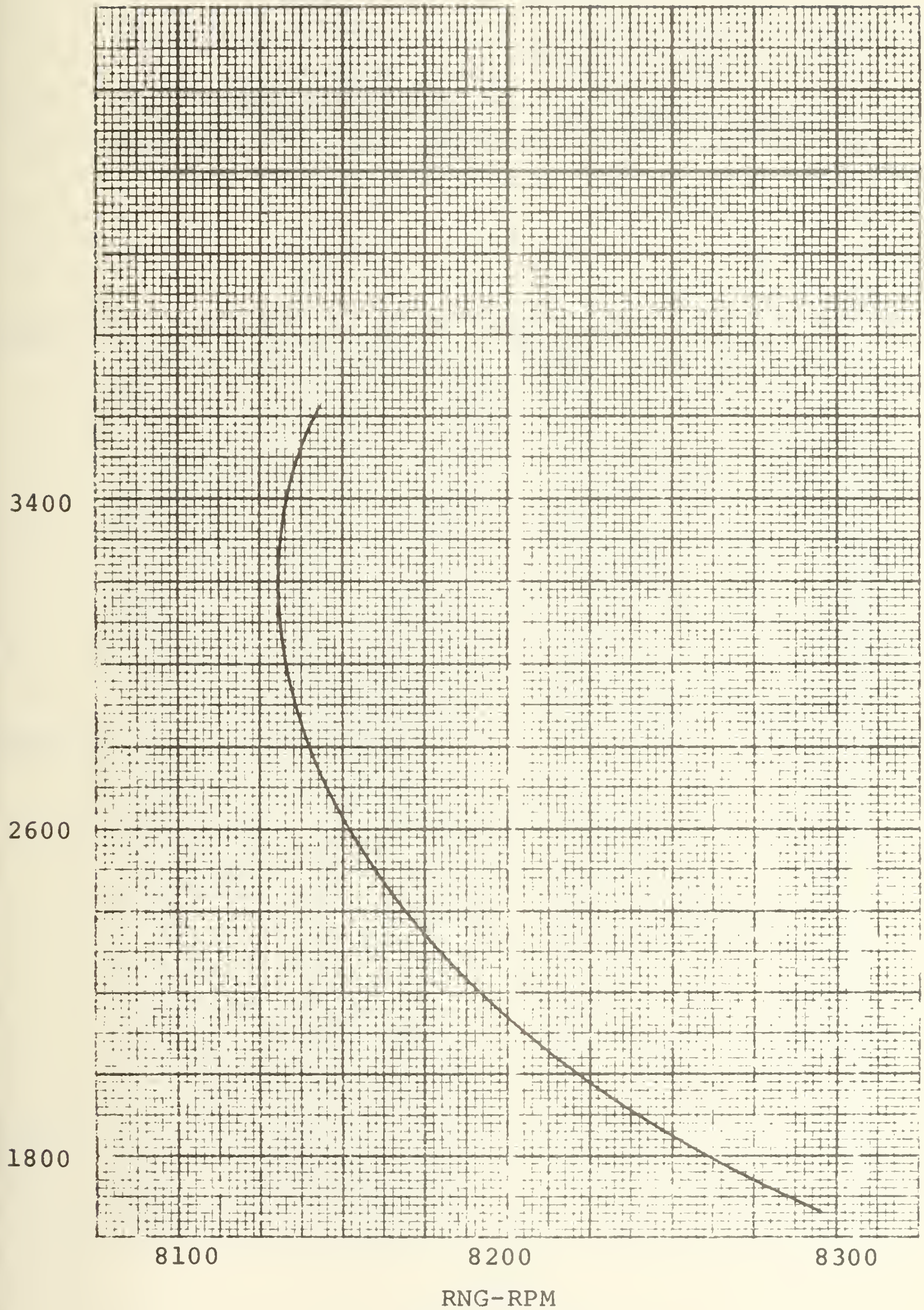


FIGURE 6-8

TORQUE vs RNG - 10000 HP - 59°F

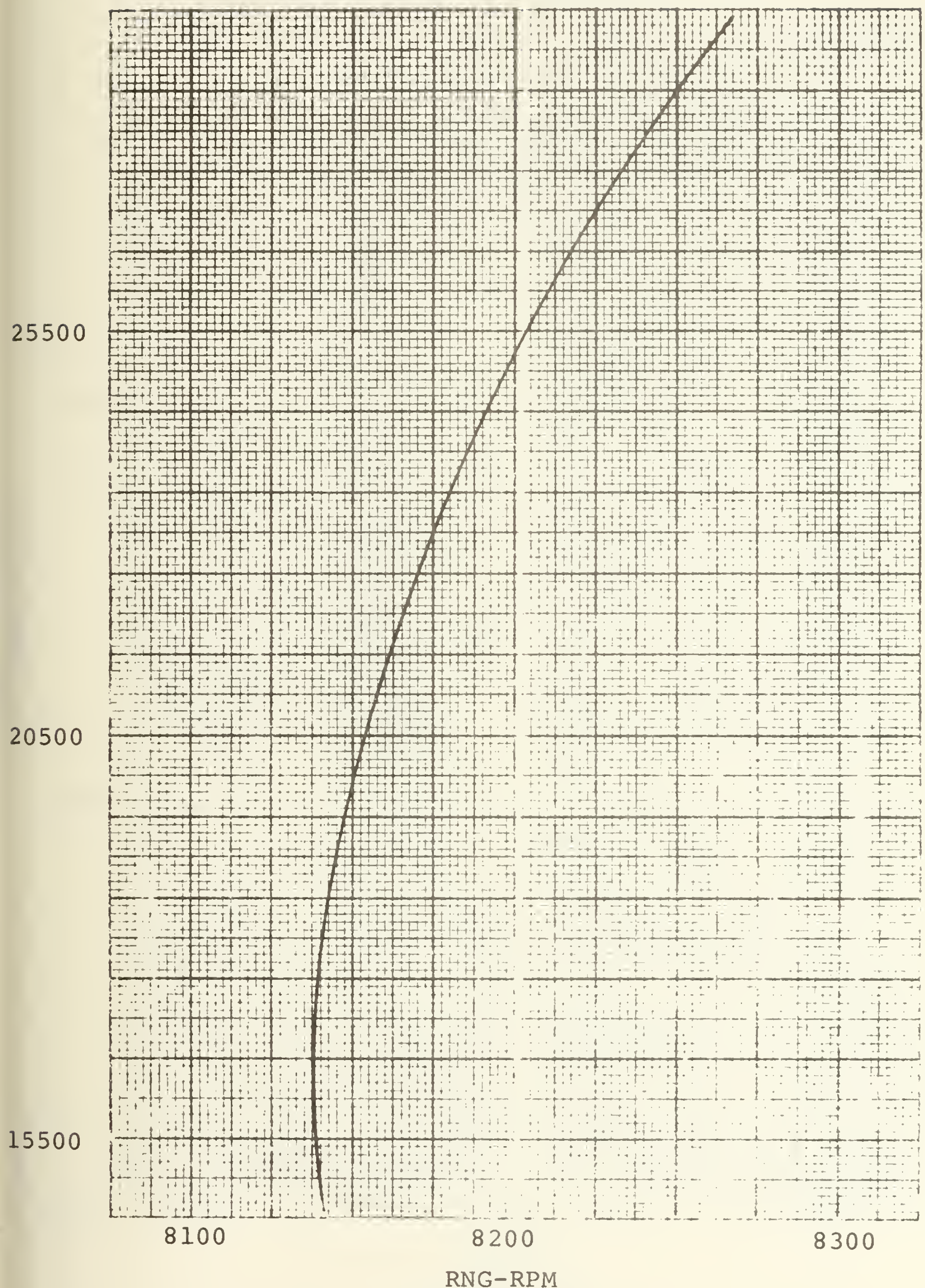


FIGURE 6-9

T054 vs RNG - 15000 HP - 59°F

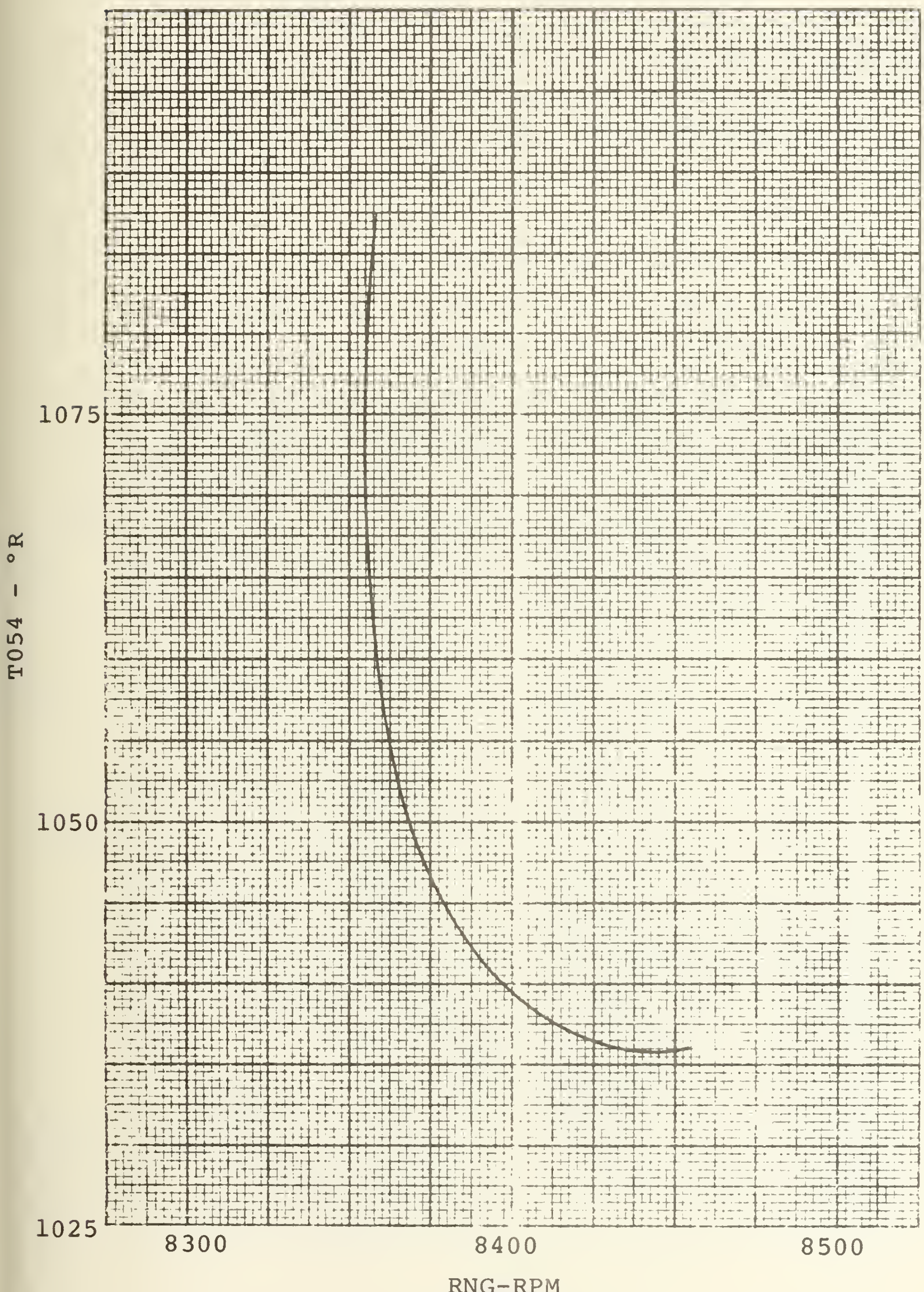


FIGURE 6-10

P054 vs RNG - 15000 HP - 59°F

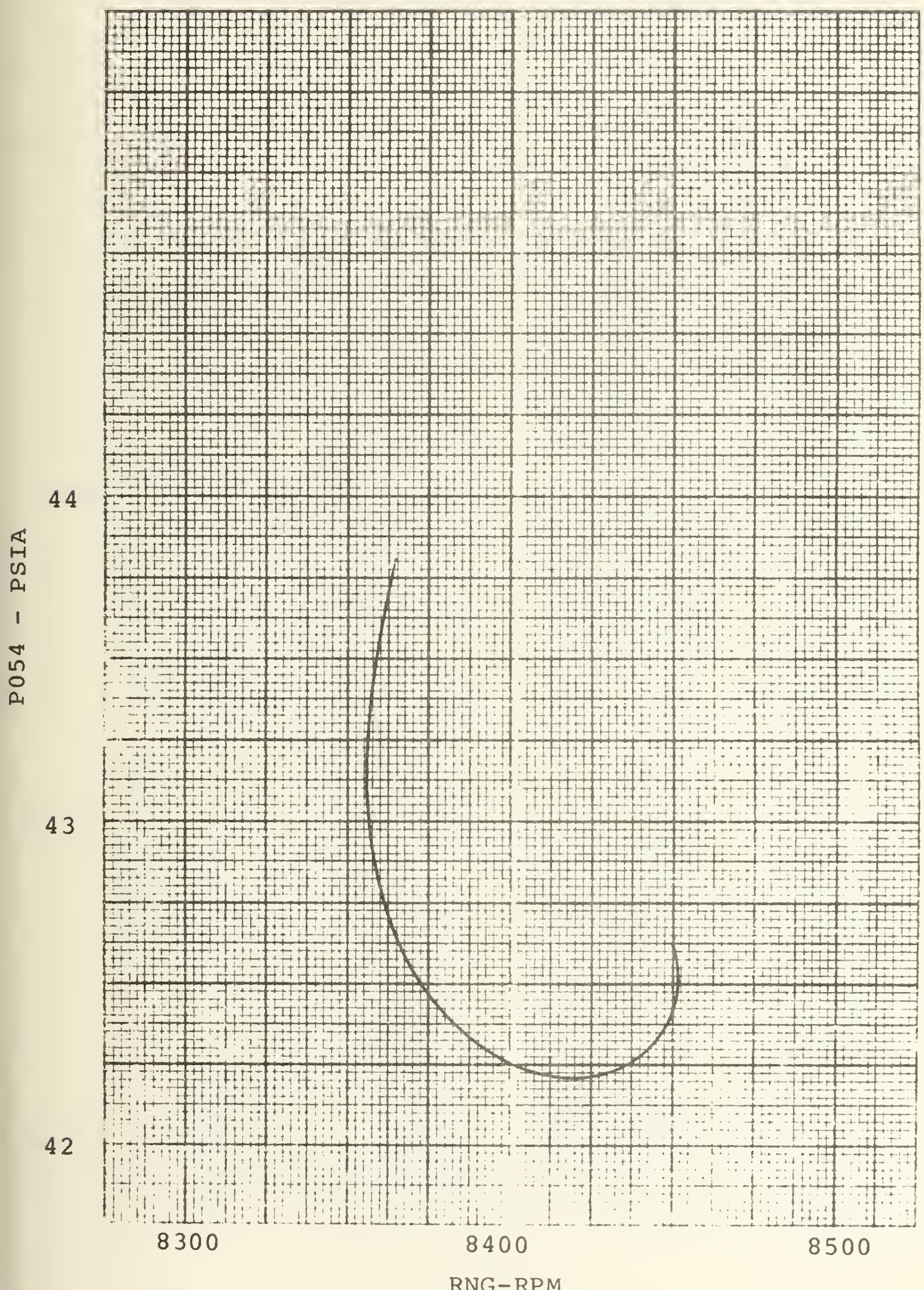


FIGURE 6-11

WF vs RNG - 15000 HP - 59°F

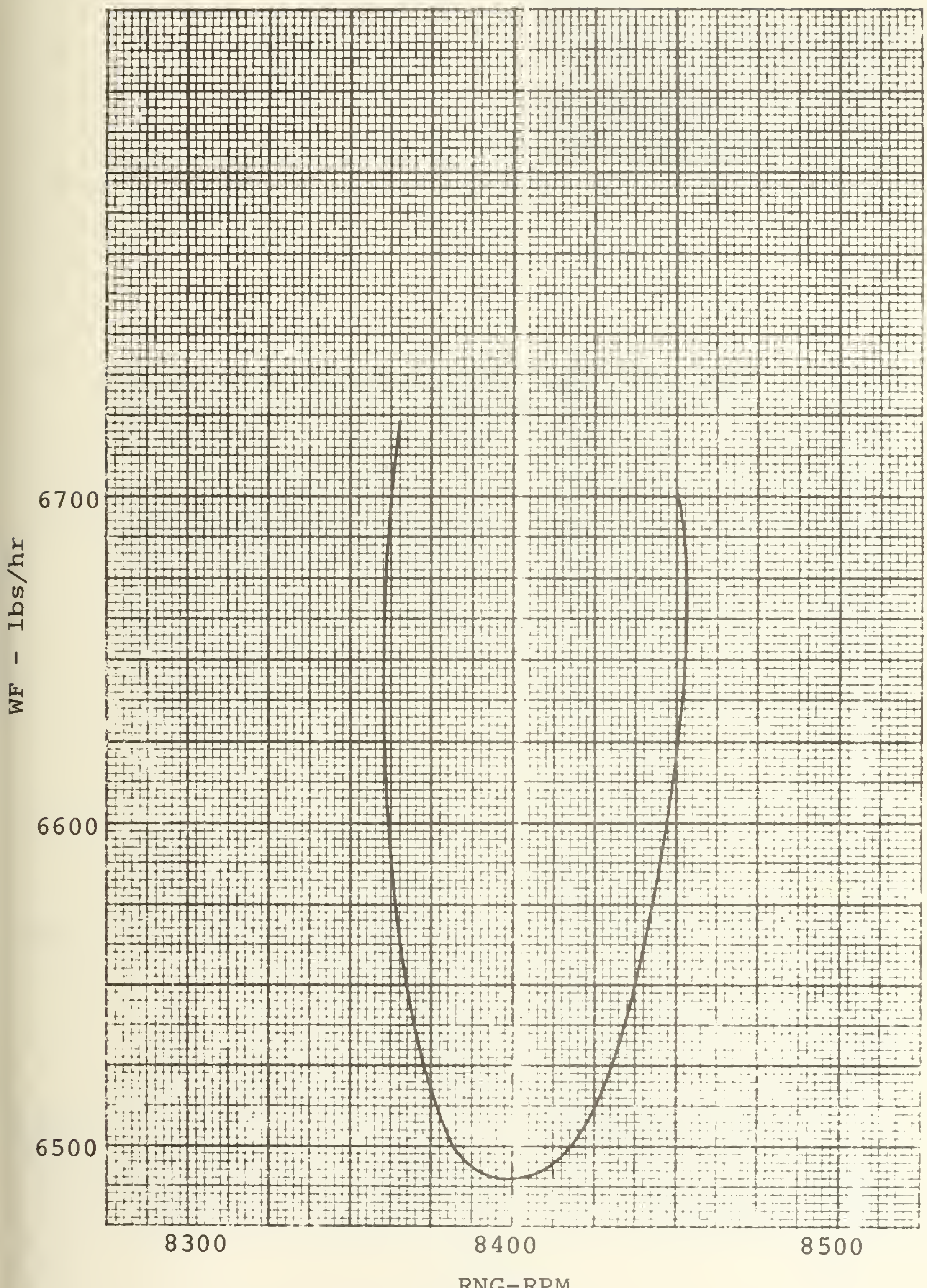


FIGURE 6-12

T054 vs RNG - 20000 HP - 59°F

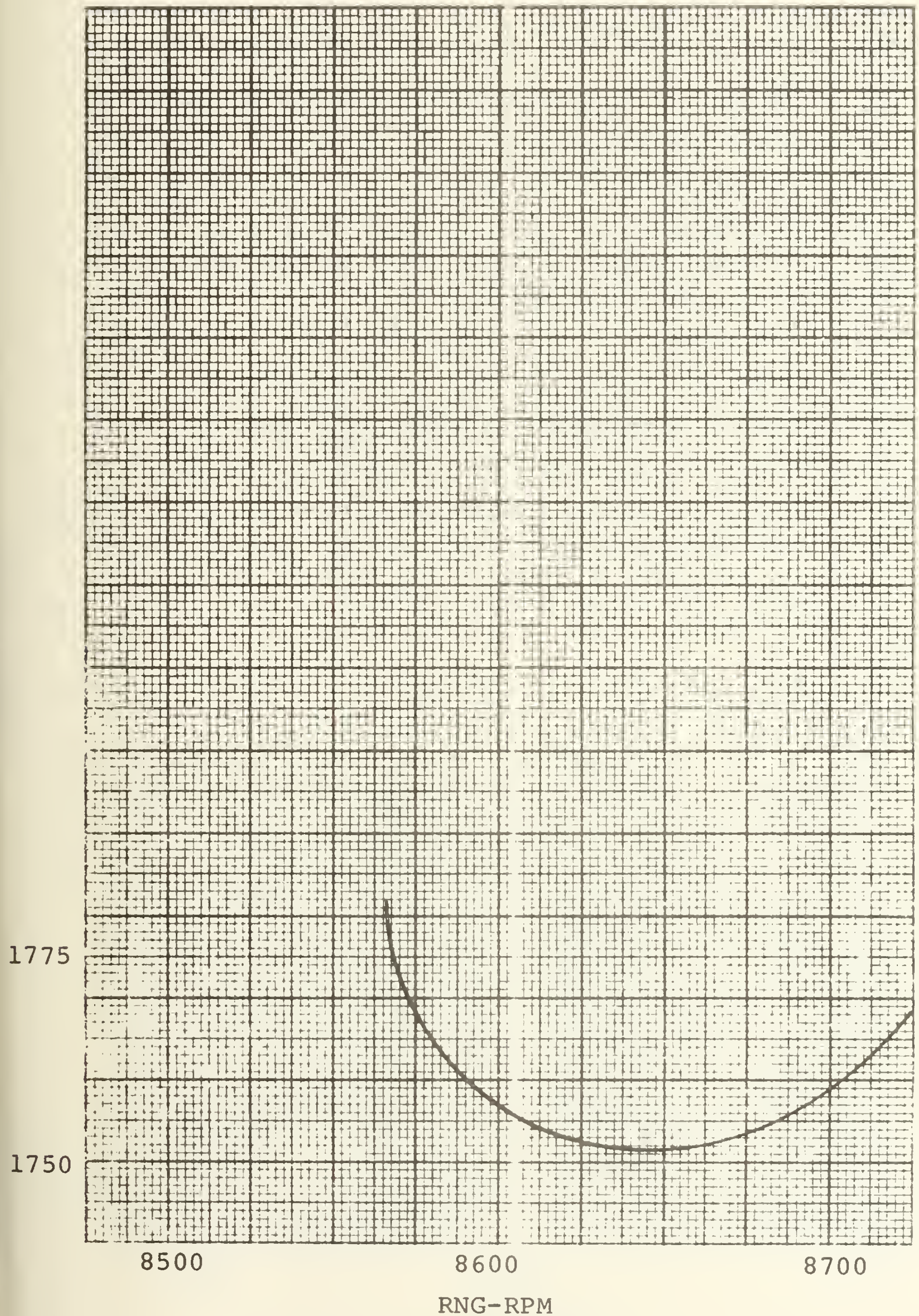


FIGURE 6-13

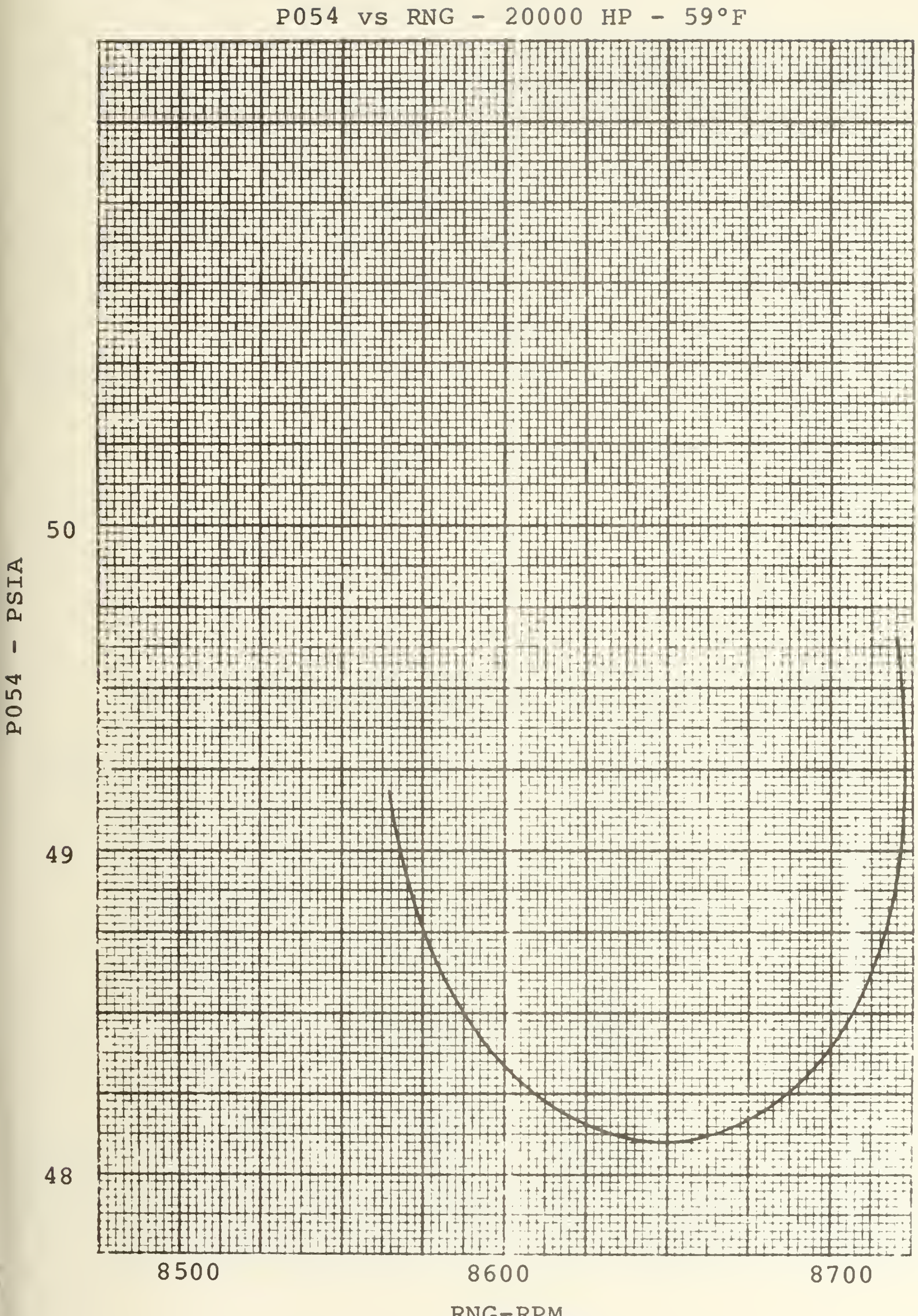

P054 vs RNG - 20000 HP - 59°F
P054 - PSIA
50
49
48
8500
8600
8700
RNG-RPM

FIGURE 6-14

WF vs RNG - 20000 HP - 59°F

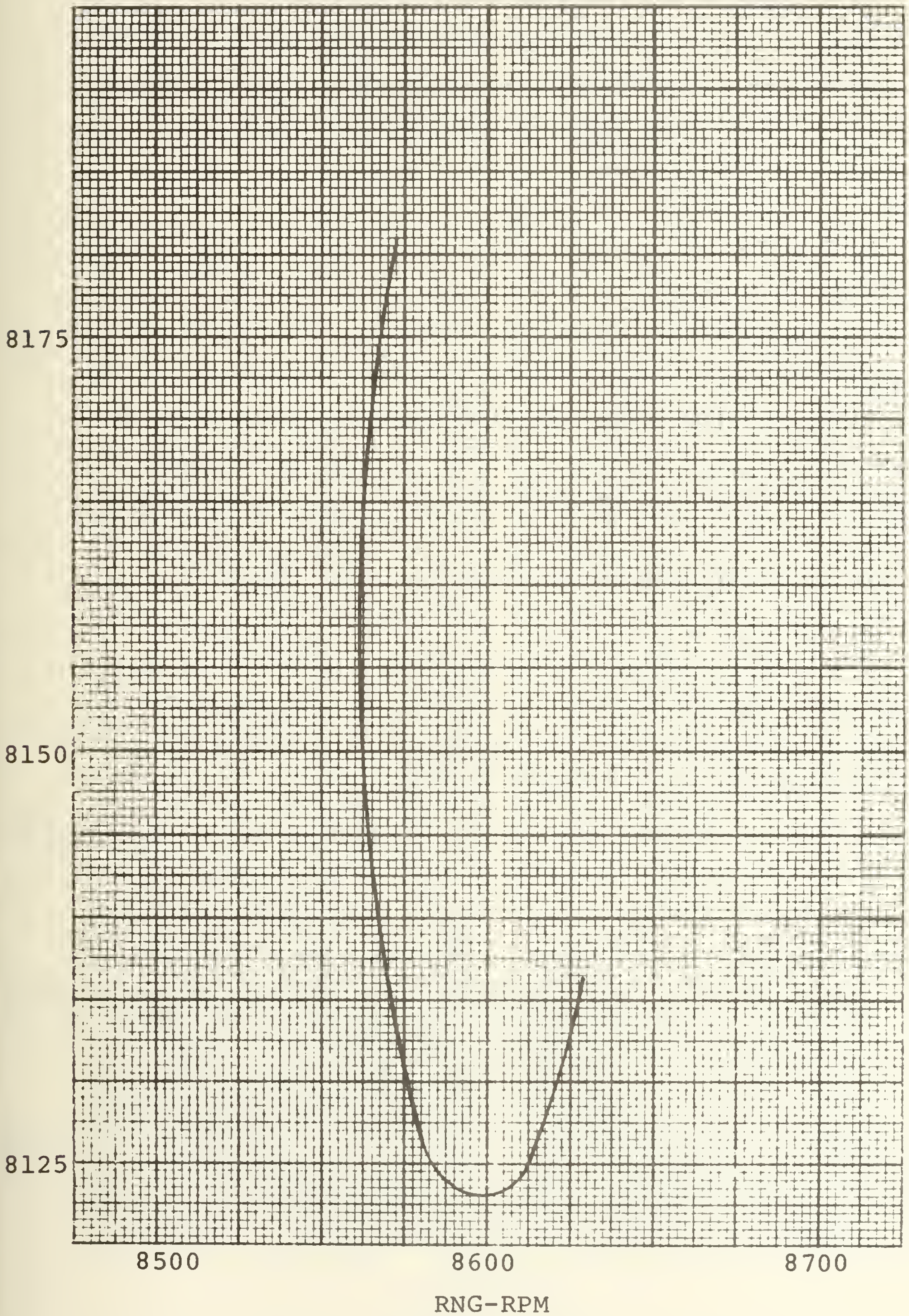


FIGURE 6-15

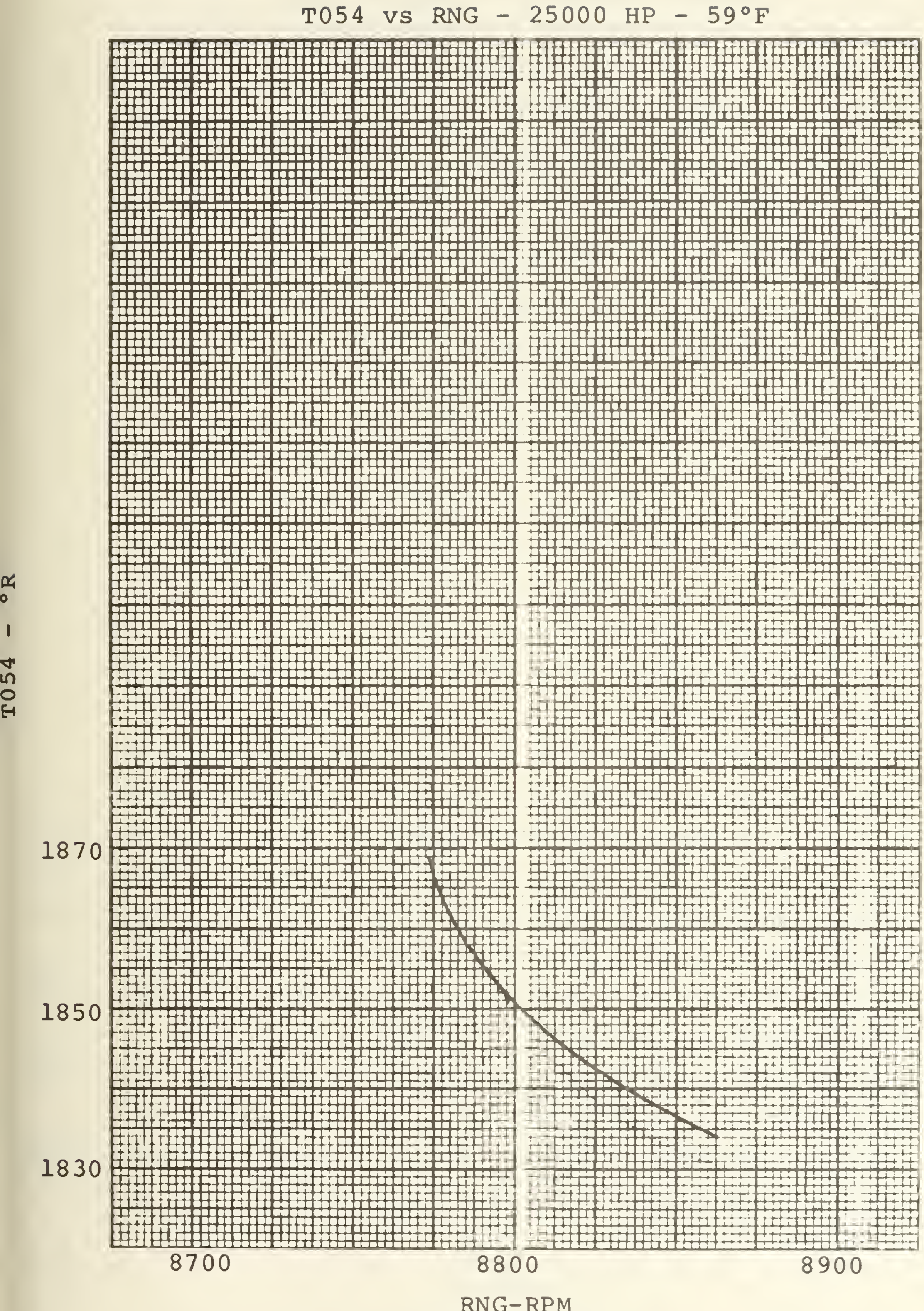

T054 vs RNG - 25000 HP - 59°F
T054 - °R
1870
1850
1830
8700
8800
8900
RNG-RPM

FIGURE 6-16

P054 vs RNG - 25000 HP - 59°F

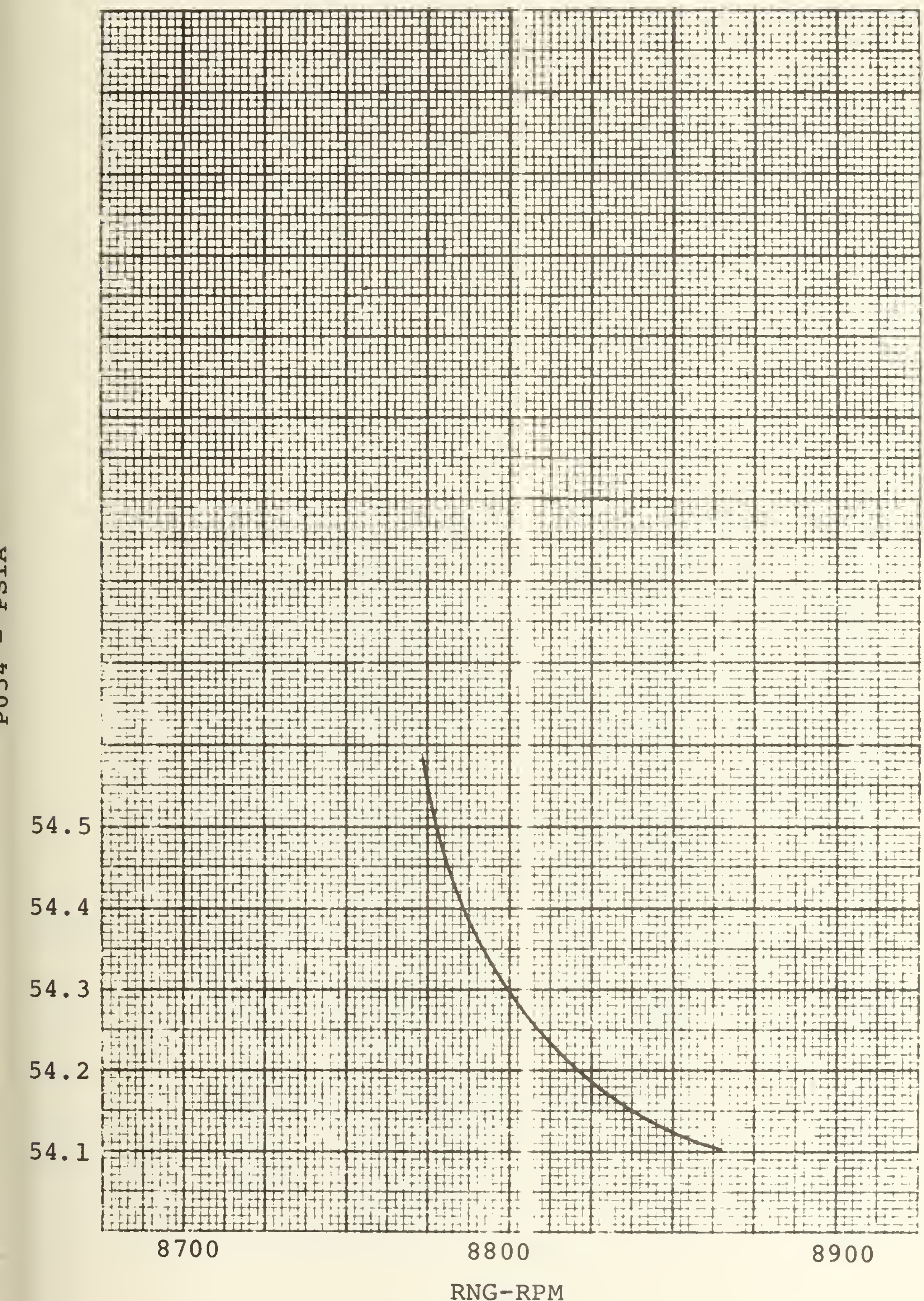


FIGURE 6-17

WF vs RNG - 25000 HP - 59°F

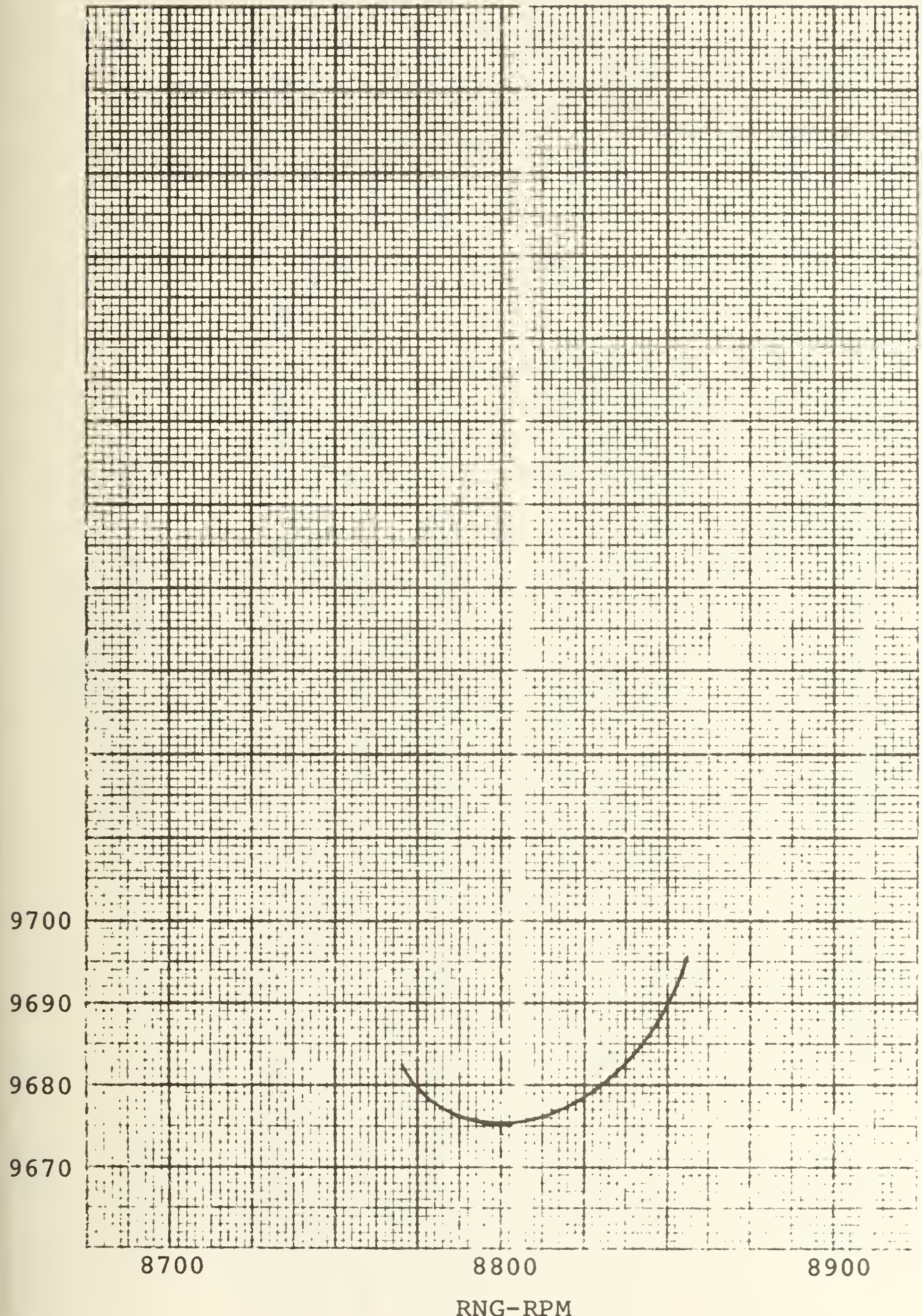


FIGURE 6-18

TABLE 6-1

GRAPHICAL BASELINE DATA

BHP	RNG	T054	PS3	PRC	P054	WF	RNPT	TORQUE	PRGG	SFC	W2	W8
10000	8200	1475	155.6	10.59	35.32	4795	2182	24533	2.40	.4804	101.5	102.0
15000	8400	1639	183.5	12.49	42.25	6490	2838	28032	2.87	.4327	113.9	114.7
20000	8600	1757	211.7	14.40	48.33	8123	3087	34091	3.29	.4062	126.5	127.5
25000	8800	1851	239.6	16.30	54.30	9675	3300	39789	3.69	.3870	138.6	140.1

HPTPR vs RNG at Equilibrium Lowest SFC 59°F

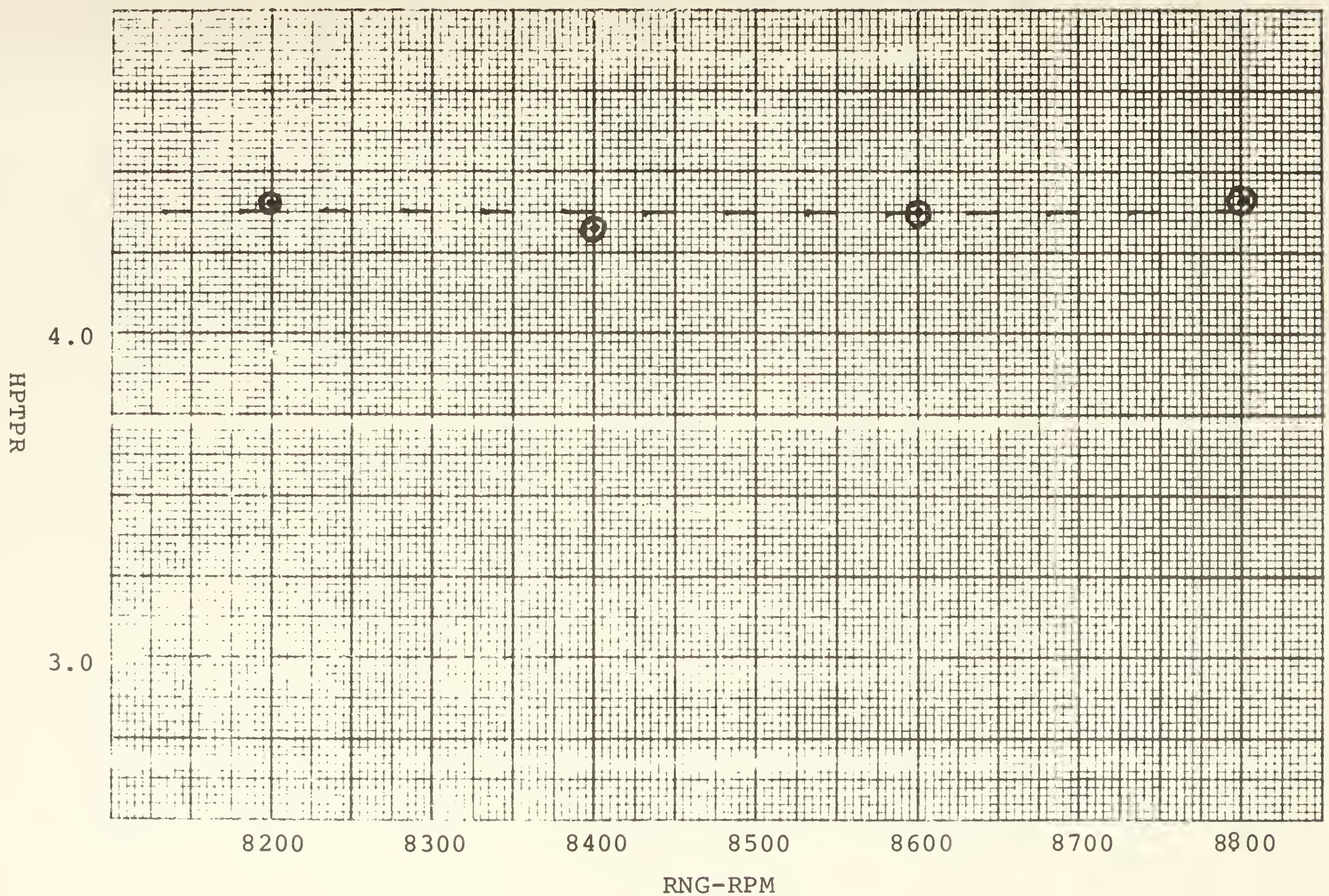


FIGURE 6-19

combustion), and the value of the pressure ratio is nearly constant; with a .8% deviation from the average. The deviation is negligible and shows that the first set of data is probably within a one percent error and that the power turbine is choked. However, using some of the theory established in Chapter IV, the data can be refined. The data for 25000 horsepower is a column from the table in Reference [1], and therefore no interpolation was necessary to find the parameters. The data sets the value of HPTPR, and using the ellipse law, Equation (6.1), it allows for an accurate value of K to be calculated, which is a constant for the machine.

$$NDF \equiv \frac{W\sqrt{T_{054}}}{P_{054}K} = \left[1 - \left(\frac{P_{05}}{P_{04}}\right)^2\right]^{1/2} \qquad (6.1)$$

From the ellipse law,

$K = 115.38$

and the data specifies

$HPTPR = 4.41$

As the right hand side of Equation (6.1) goes to one, the

power turbine becomes choked. Substituting the values of parameters and K into the left hand side yields a value slightly less than one (Equation 6.2).

$$\frac{W\sqrt{T_{054}}}{P_{054}K} = .9621 \qquad (6.2)$$

But the equation is developed for the nozzle area of the turbine, and the values of T_{054} and P_{054} are obtained prior to the turbine inlet. The value of Equation (6.2) is as stated and constant for minimum SFC operation. Using K as found, the right hand side of Equation (6.1) was evaluated with the data from each column of Reference [1] for 59°F and for the values of Table 6-1. The outcome is tabulated in Table 6-2 below.

TABLE 6-2

NDF FOR EACH COLUMN

HP \ COL	1	2	3	4	5	6	*
10000	1.0008	.9695	.9360	.9052	.8929	.8821	.9609
15000	.9969	.9713	.9412	.9269	.9147	---	.9526
20000	.9875	.9626	.9487	.9355	---	---	.9589
25000	.9751	.9620	.9493	---	---	---	.9621

* Value for data from Table 6-1.

The only value of 1.0 obtained is from the first column of the 10,000 horsepower data, and the SFC in this column is 0.6865; way out of limits. It can be concluded that the power turbine is operating very near a choked condition. It appears that the choked condition corresponds to a value of NDF of about 0.9621, and for values above this, the SFC increases rapidly, indicating that the turbine may be going into a supersonic flow regime. The values continue to increase because of the uncertainty and position of the measurement. Below this value, the turbine is no longer choked. Considering the value for the column in Table 6-2 corresponding to the most economical operating point, Table 6-1, and the graph in Figure 6-19, the data arrived at for 15000 HP is not good, and the values for 10,000 and 20,000 horsepower are closer. The values of PS3 in each case should be very close since the pressure is a linear function of speed. This is illustrated in Figure 6-20, which is a plot of gas generator speed versus pressure ratio across the compressor. The data is from each column in the data in the appendix. The line is disjointed because each segment represents a distinct horsepower, and each is non-linear at the lower end. The non-linearity is a result of extreme lower operational limits of the gas generator with an unchoked power turbine. The overall effect is a straight

PRC RANGE vs RNG - 59°F

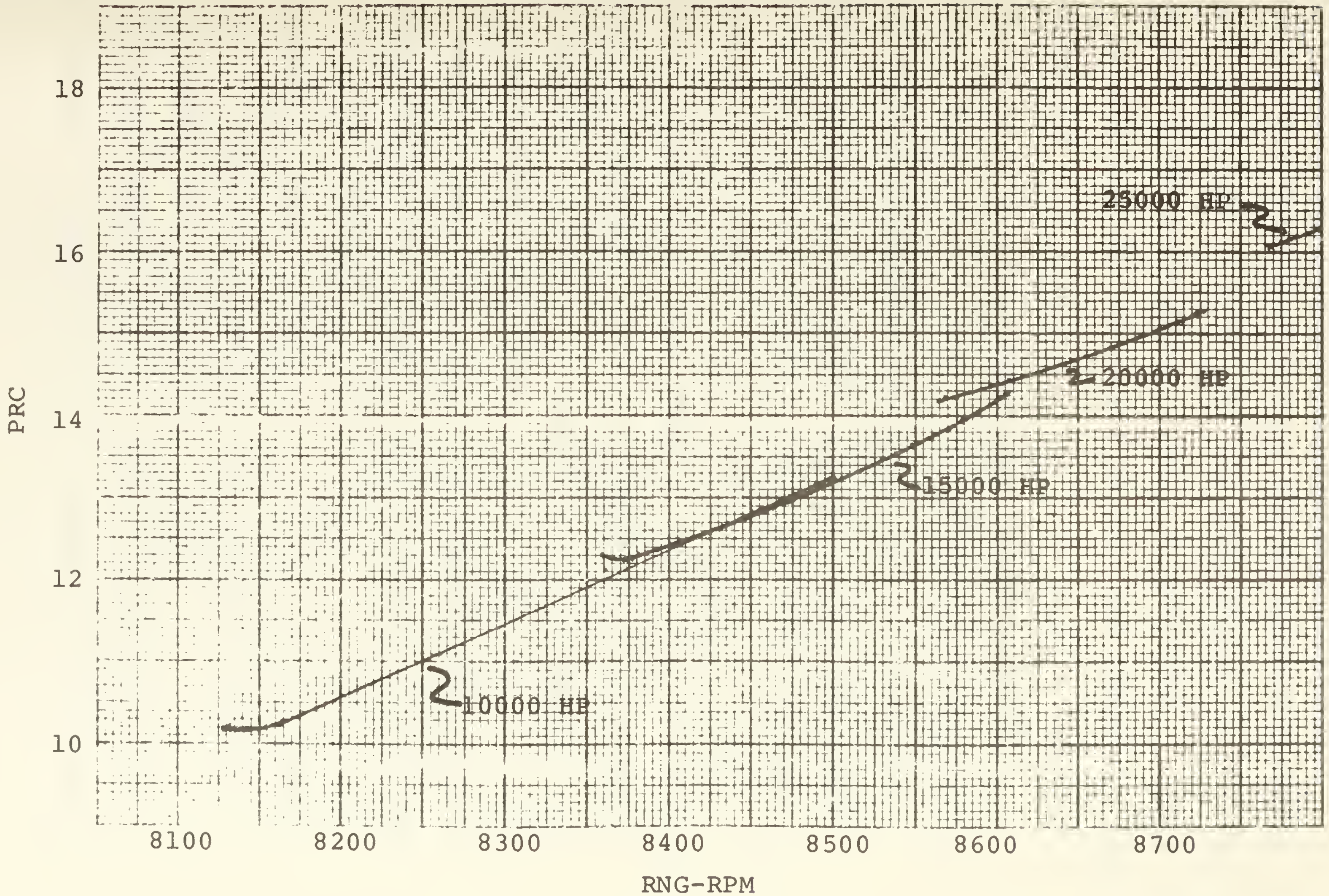


FIGURE 6-20

line, however, which supports each value of PS3 in Table 6-1. So, the value of P_{054} can be altered slightly to force the value of HPTPR for all horsepowers to be equal to that of 25,000 horsepower, 4.41. In addition, a range can be established for the value of NDF, and therefore T_{054} can change and vary slightly. The flow rate is accurate because it is linear with compressor speed and K is a constant. When this is done, the new value of T_{054} corresponding to most economical operation must lie near the original line in the graphical plot or form a new smooth line. In the case of 15,000 horsepower, the value of T_{054} was altered by 16° when a new value of P_{054} was chosen. Looking at Figure 6-10, the shape of the curve in the 8400 RPM range is uncertain, so a 15° change could be possible. It is difficult to determine the shape of the graph from four points because the minimum is not specified. However, in the case of 20000 horsepower, the temperature could not be changed because the shape of the curve was established in the area of 8600 RPM's (Figure 6-12). The resulting parameters are shown in Table 6-3. These are the values that can be plotted against the gas generator speed and then programmed into the computer module.

Included in Table 6-3 are several parameters which warrant further discussion and are results of calculations

TABLE 6-3

FINAL BASELINE DATA

BHP	RNG	PS3	PRC	η_c	TS3	W2	WF	T_{04}	T_{054}	P_{054}	PRGG	η_{HTP}
10000	8200	155.6	10.59	85.59	1102	101.5	4795	2090	1472	35.28	2.40	85.60
15000	8400	183.5	12.49	85.88	1157	113.9	6490	2291	1619	41.60	2.83	85.00
20000	8600	211.7	14.40	86.36	1205	126.5	8123	2478	1757	48.00	3.27	84.25
25000	8800	239.6	16.30	87.12	1245	138.6	9675	2609	1851	54.30	3.69	84.05

BHP	W8	CP8	SFC	RNPT	TORQUE	HPTPR	NDF	T04/T054
10000	102.0	.2645	.4804	2182	24533	4.41	.9614	1.42
15000	114.7	.2680	.4327	2838	28032	4.41	.9615	1.42
20000	127.5	.2710	.4062	3087	34091	4.41	.9650	1.42
25000	139.4	.2736	.3870	3300	39789	4.41	.9620	1.42

using the parameters established in this chapter. The compressor discharge temperature is found graphically as discussed previously (graphs not included), and the total to static enthalpic efficiency is then calculated from Equation (6.3), assuming $\gamma = 1.4$.

$$\eta_c = \frac{T_2[(P_{S3}/P_{02})^{\gamma-1/\gamma} - 1]}{\Delta T_{S3-2}} \tag{6.3}$$

The efficiency of the high pressure turbine and the combustor exit temperature are found through an iteration procedure, using Equations (6.4) and (6.5) which are derivatives of equations in Chapter IV and based on the fact that gas generator turbine work equals compressor work.

$$\Delta T_{454} = \frac{Cp\Delta T_{S32\ actual}}{\eta_{HPT} Cp8} \tag{6.4}$$

$$\eta_{HPT} = \frac{\Delta T_{454}}{T_{03}[1 - (P_4/P_3)^{\gamma-1/\gamma}]} \tag{6.5}$$

If the static to total enthalpic efficiency of the turbine is first guessed and the change in temperature across the turbine calculated, a new efficiency can then be calculated. The value of CP8 in the table is a result of linear

interpolation. As stated previously, the values of component efficiency change only slightly throughout the range of operation. Additionally, the combustor exit temperature was calculated to illustrate that the ratio of T04/T054 is a constant as it should be for choked operation of the power turbine. This temperature was also calculated for several columns of the data at random, and the result was that the variance of the ratio was very slight. For this reason, it was not considered a useful parameter for monitoring the engine. The pressure ratio across the high pressure turbine did vary considerably, and therefore it is useful as a monitoring indicator. From Equation (4.14),

$$\frac{T_{054}}{T_{04}} \times \frac{P_{04}}{P_{054}} = \frac{A_4}{A_3} \tag{4.14}$$

if the temperature ratio varies slightly, the pressure ratio will more adequately reflect the change in areas as the nozzles bow or erode. The ratio will remain nearly constant because as the turbine efficiency drops, more fuel is required. Thus, the combustor exit temperature increases with increasing turbine exit temperature. Figure 6-21 illustrates how the pressure ratio across the high pressure turbine changes at 59°F as the compressor pressure ratio changes for a constant horsepower. Consequently, there are more

HPTPR RANGE vs PRC - 59°F

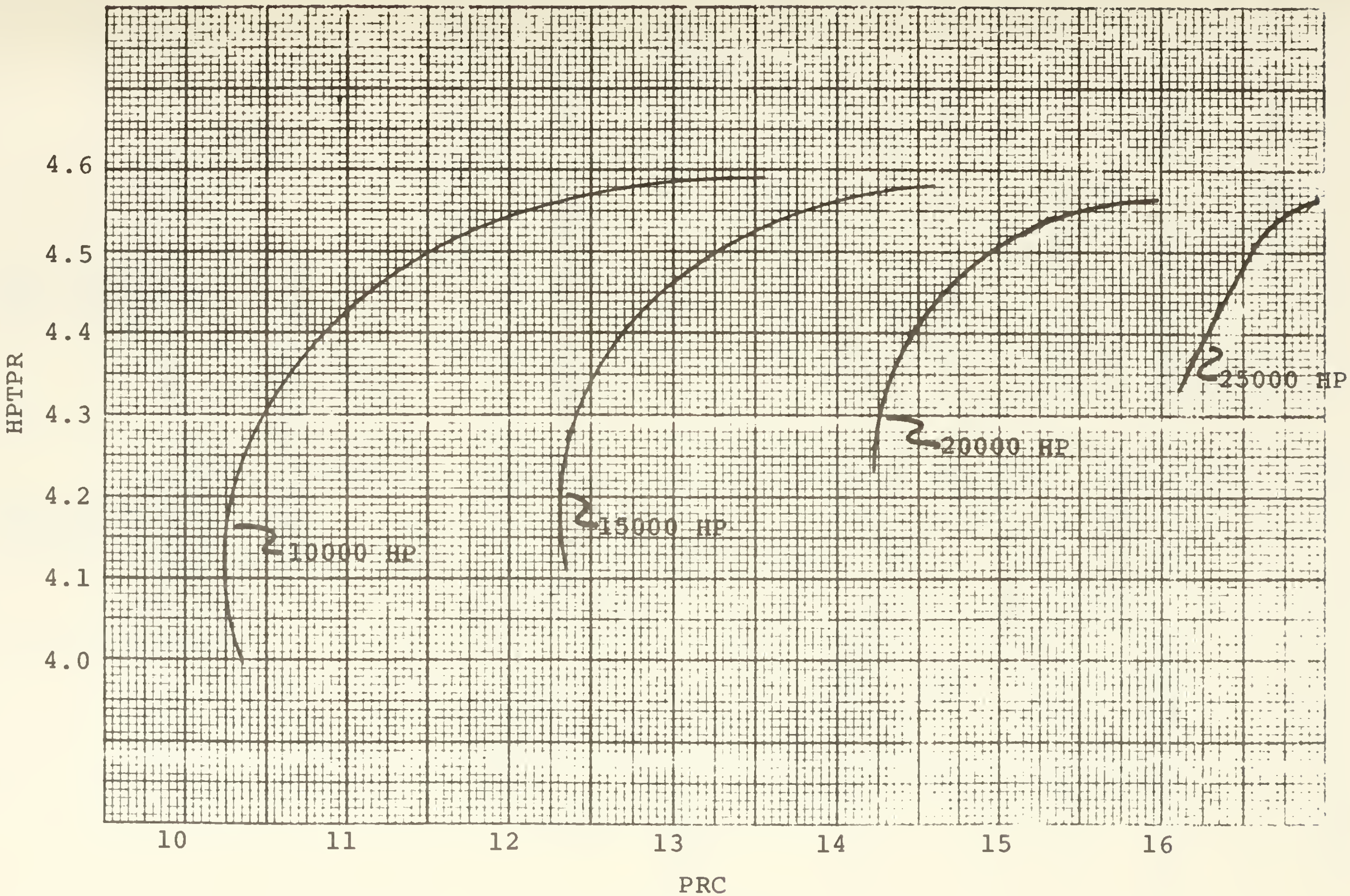


FIGURE 6-21

parameters listed in Table 6-3 than the monitoring procedure dictates, but they serve to illustrate the validity of the thermodynamic analysis of the engine.

Previously it was mentioned that for maximum efficiency both turbines should run near a choked condition. The operation of the power turbine has been discussed, but the discussion of the high pressure turbine remains. Utilizing Equation (4.11) and the data for 25000 horsepower, the turbine is choked. In fact, the gas generator turbine remains choked over the complete range of operation in the columns of the appendix.

$$\frac{W8\sqrt{T_{04}}}{.523P_{04}} = A_3 \qquad (6.6)$$

With the 25000 horsepower data used to solve for A_3, the remainder of the data can be checked, and the results are tabulated in Table 6-4.

TABLE 6-4

AREA OF HP TURBINE

HP	AREA (ft^2)
10000	.3965
15000	.3971
20000	.3950
25000	.3924

Since the areas calculated are approximately equal, the data for each horsepower level is accurate. Finally, if the pressure ratio of the turbine is considered, Table 4-1 yields a value of 1.0, indicating that the machine is choked. From the data the pressure ratio in each column is near four, so that the gas generator is always choked.

It has been established that the data in Table 6-3 corresponds to the most economical points of operation for the engine, but it is not in a very useful form for the computer module. Plots were made of the data required for monitoring versus the independent variable, RNG (Figures 6-22 through 6-27). Then, equations for the curves were found utilizing a least squares curve fitting Fortran IV routine, and are included in Table 6-4. It is the equations which are programmed into the computer module.

PRC vs RNG - Lowest SFC - 59°F

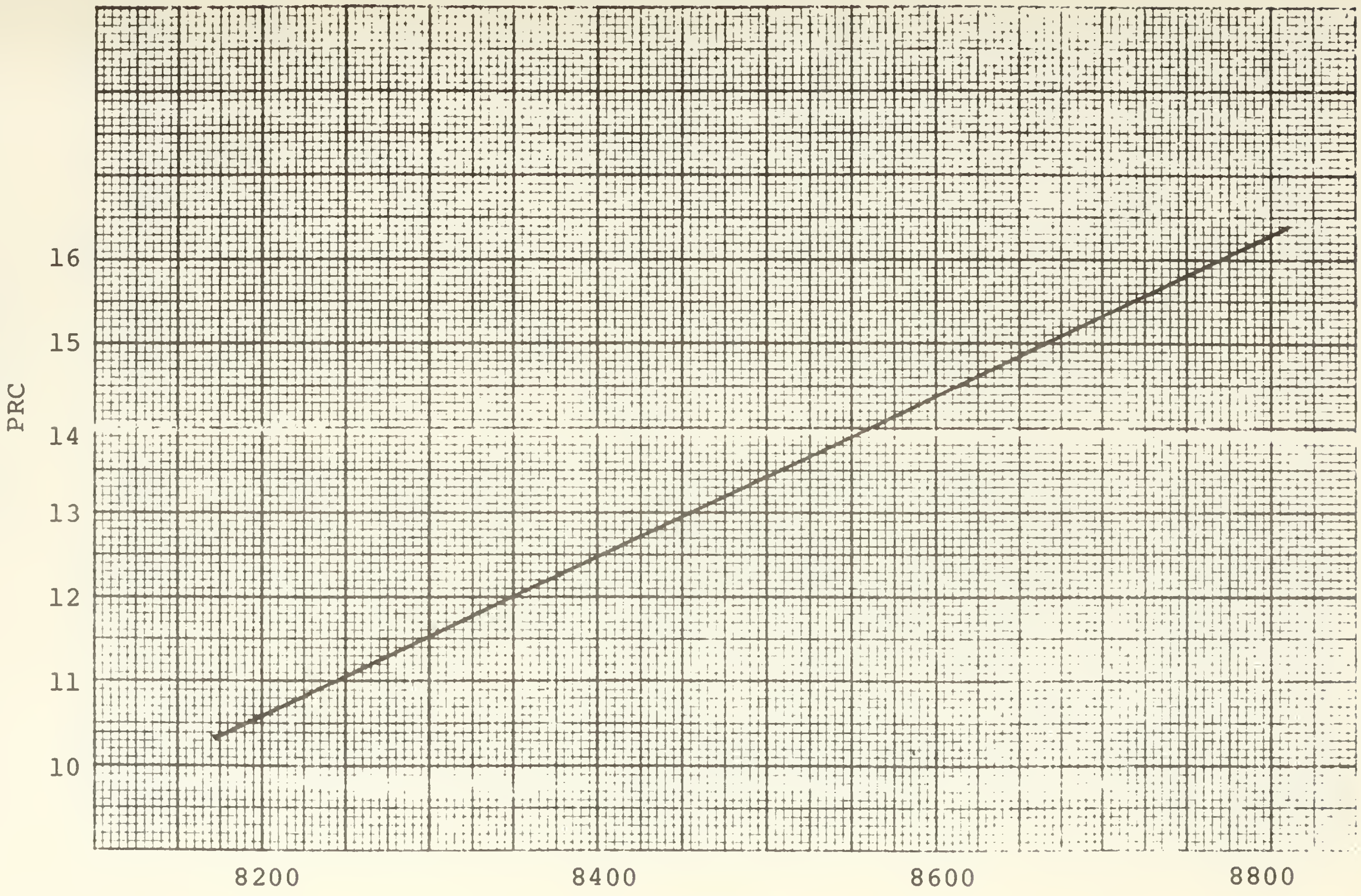


FIGURE 6-22

SFC vs RNG - Lowest SFC - 59°F

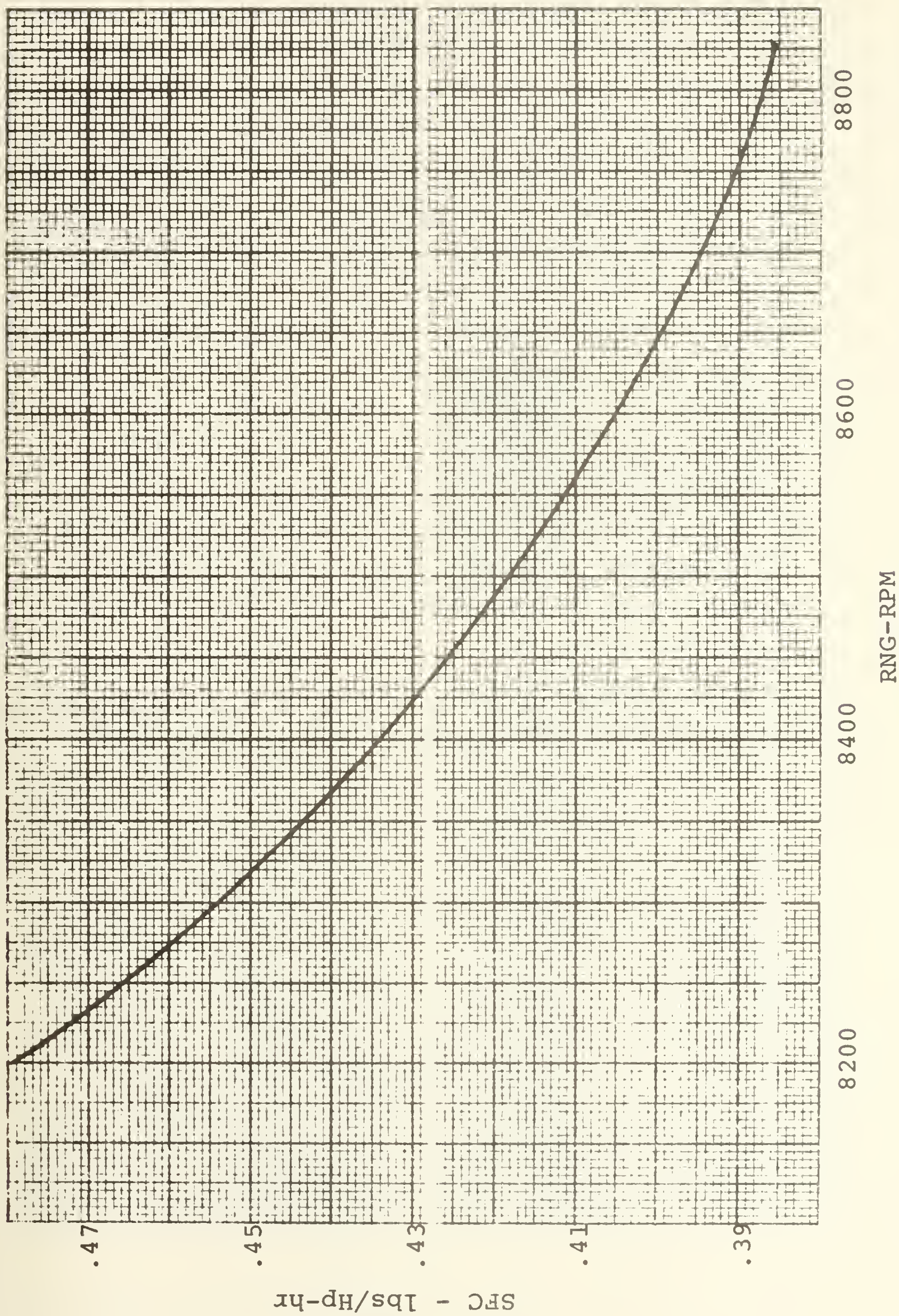


FIGURE 6-23

HPTPR vs RNG - Lowest SFC - 59°F

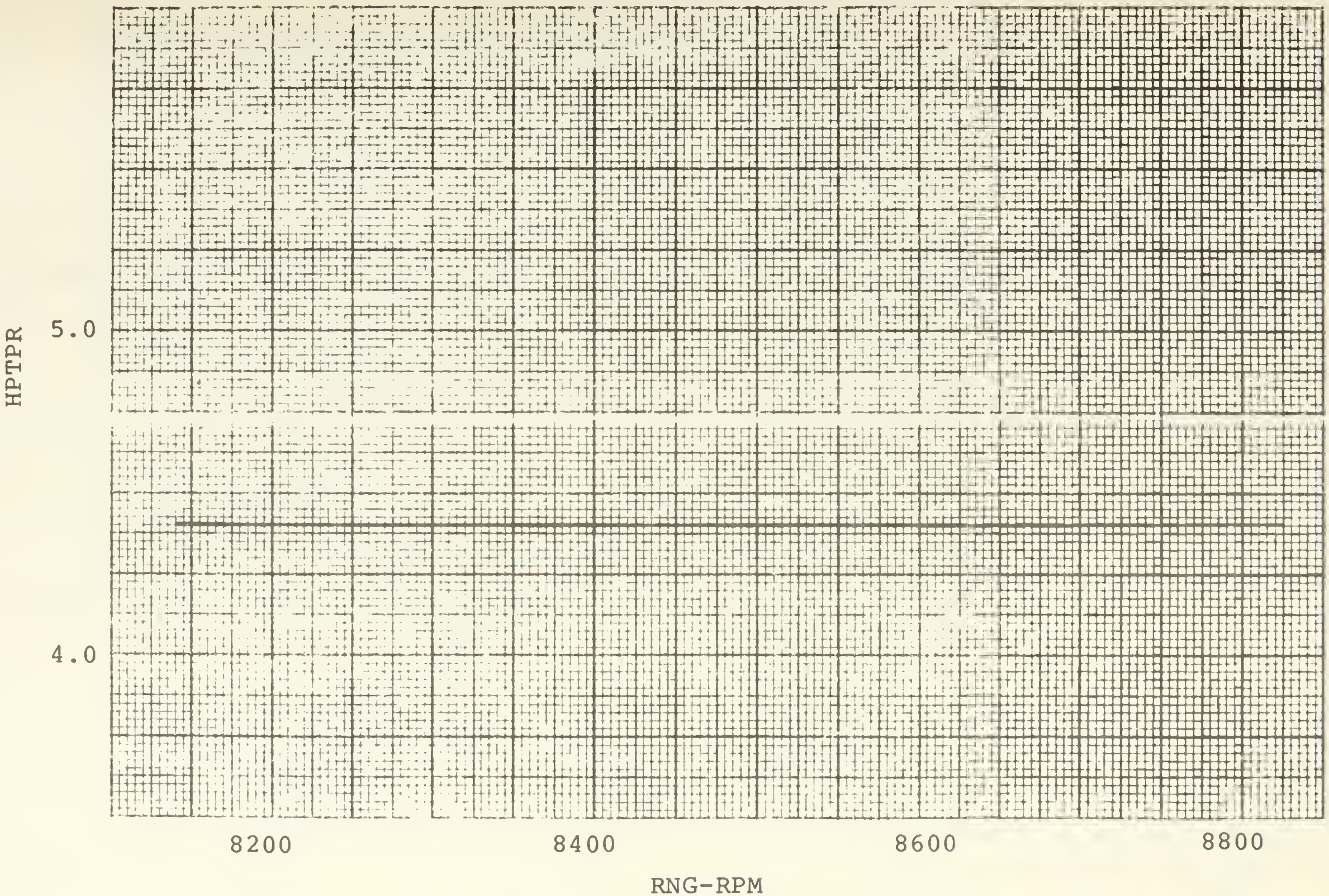


FIGURE 6-24

T054 vs RNG - Lowest SFC - 59°F

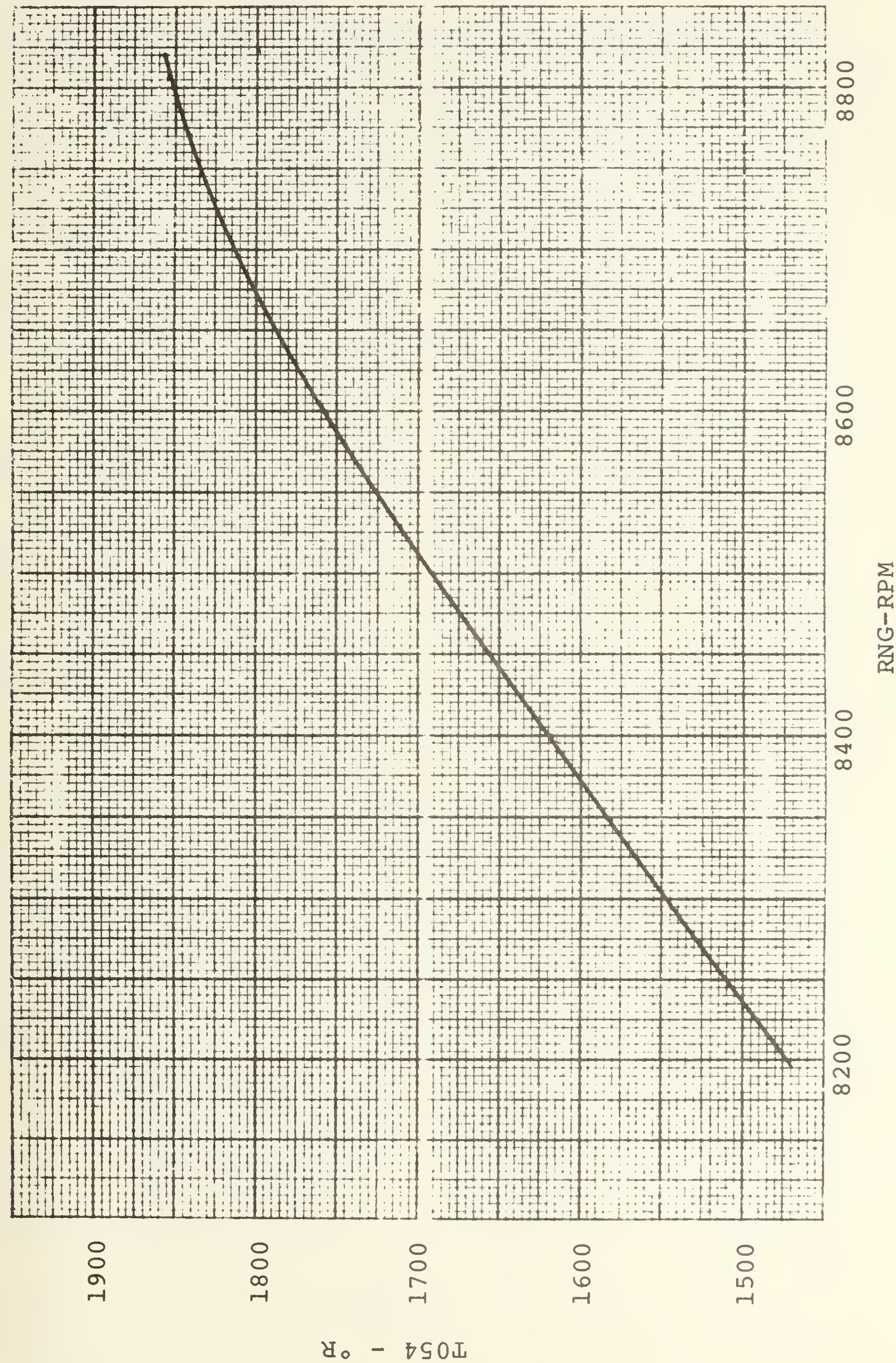


FIGURE 6-25

PRGG vs RNG - Lowest SFC - 59°F

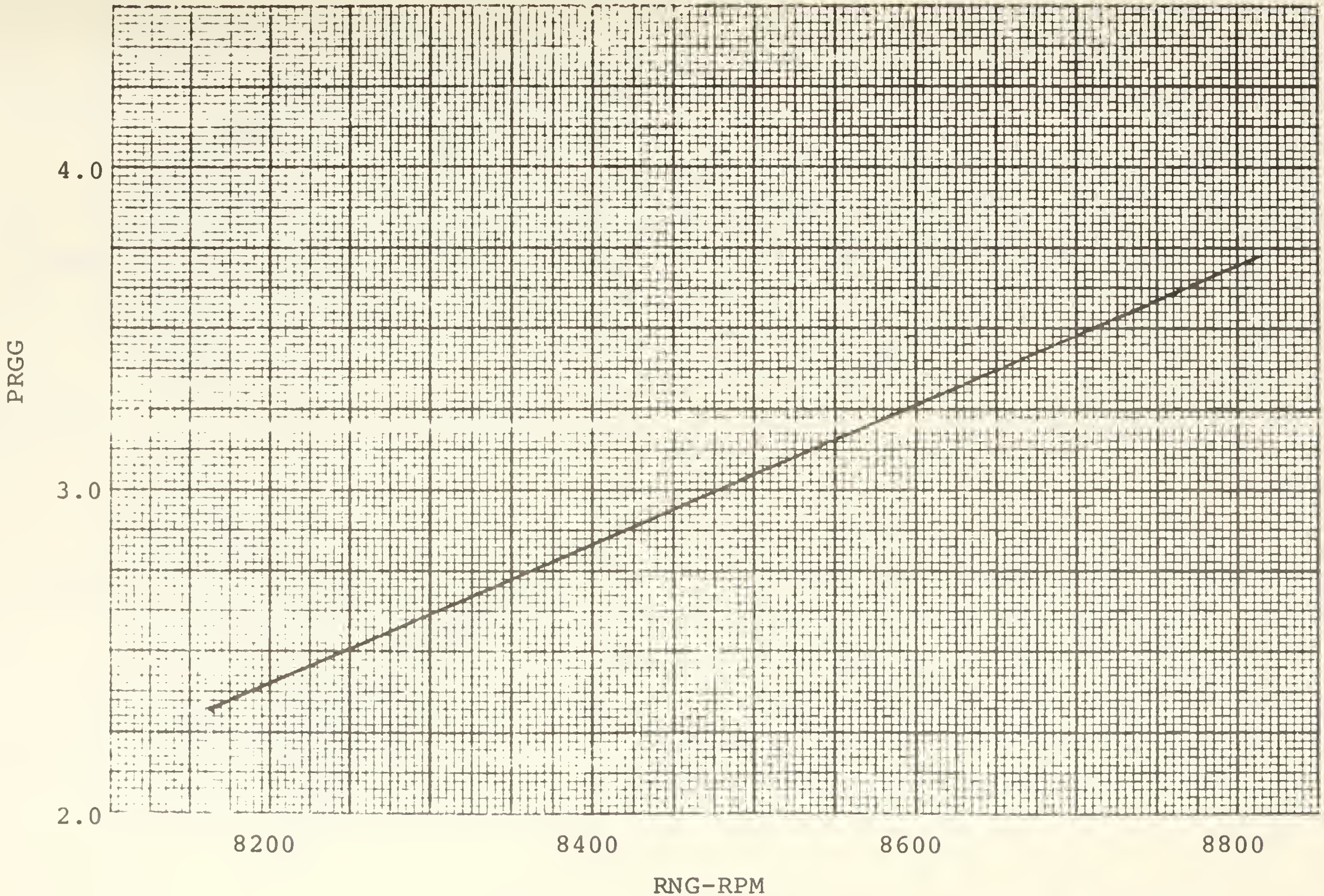


FIGURE 6-26

BHP vs RPM - Lowest SFC - 59°F

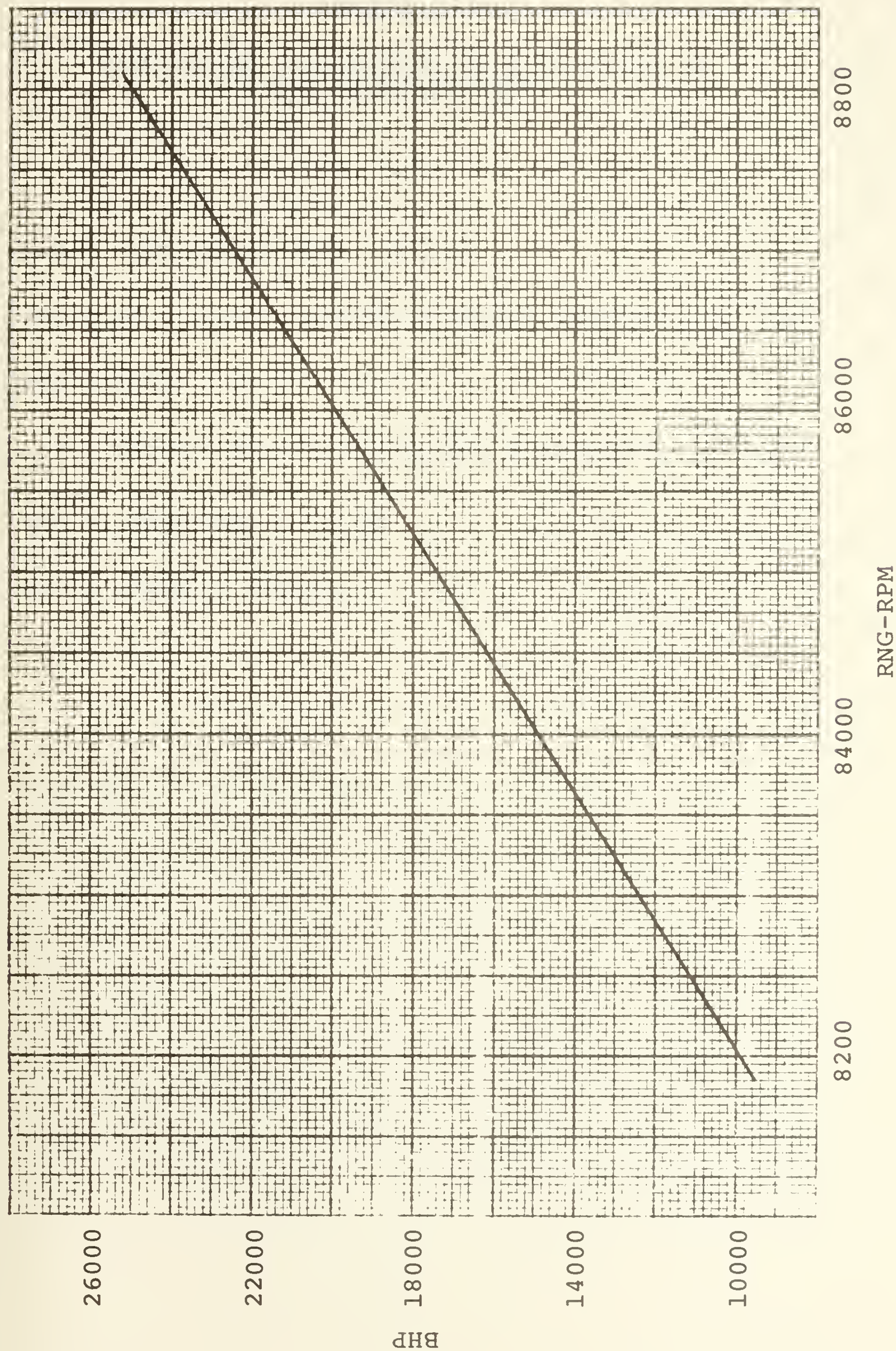


FIGURE 6-27

TABLE 6-5

MONITORING PARAMETERS AS A FUNCTION OF GAS GENERATOR RPM

$$PRC = .0095166667RNG - 67.43$$

$$SFC = 20.417603 - .0045577921RNG + 2.5935799 \times 10^{-7} RNG^2$$

$$HPTPR = 4.41$$

$$T054 = -34993.883 + 7.9785061RNG - 4.3082377 \times 10^{-4} RNG^2$$

$$PRGG = .00215RNG - 15.23$$

$$BHP = 25RNG - 195000$$

CHAPTER VII

CORRECTIONS

A variation in ambient conditions causes the output of the gas turbine engine to change. As the ambient temperature decreases, the output of the machine increases for the same gas generator speed. This is because it requires less work to compress cooler air, so the pressure ratios will be increased, meaning more power out. As the ambient pressure increases, the output increases because the increase in pressure has increased the density of the fluid and therefore the mass flow also. In addition to the atmospheric effects, the pressure losses in the inlet and exhaust ducts cause a decrease in power. The loss in the inlet duct decreases the pressure and thereby the mass flow; and the loss in the exhaust duct puts a back pressure on the engine. Consequently, the parameters measured at the engine have to be corrected, using standard gas turbine corrections, to baseline conditions, International Standard Organization Day, before they can be compared if the trending is to be of any value.

Ambient temperature has the greatest effect on the operation of the engine. Standard gas turbine correction procedure is to divide the temperatures monitored by a normalizing factor. The factor is designated theta and

raised to a particular power depending on the position of the temperature probe in the engine. The standard ambient temperature correction for T054 is given in Equation (7.1) below.

$$T054M/THETA2^{.85} = T054C \tag{7.1}$$

where

$$THETA2 = T02/518.7$$

The "2" in "THETA" designates that the ambient temperature is taken at the compressor inlet, and the subscript "M" designates the measured value. Other engine parameters are affected by temperature including fuel flow, torque, and RPM. RPM is corrected by Equation (7.2).

$$RNGM/THETA2^{.5} = RNG \tag{7.2}$$

Parameters such as fuel flow and horsepower can be corrected in a manner similar to Equations (7.1) and (7.2), but graphs are often available which have combined the temperature effects of the parameters. For example, torque and power turbine RPM can be corrected and then the horsepower found.

Or the horsepower can be calculated and then the graph, which represents a product of the correction factors, is used to correct BHP. For the temperature corrections to SFC and BHP, Figure 7-1 is applied.

Other factors that have significant effects on the engine are the inlet and exhaust duct losses. They affect temperatures, RPM's, the fuel consumption, and the horsepower production. The loss factor is presented in the form of a graph. The graphs used for the corrections of T054, RNG, SFC, and BHP are shown in Figures 7-2 through 7-5.

Other factors that affect the output of the engine are ambient pressure and altitude, humidity, and bleed derivatives. Normally for a marine gas turbine engine, the ambient pressure is ignored because sea level pressure variations are insignificant. Humidity has a negligible effect on the engine, but bleeds can be significant. Corrections for bleeds involve the measurement of mass flows and bleed flows, but if the bleed is secured, then there is no need to consider a correction factor. The module is designed without considering auxilliary bleeds and so will not print reliable data when the bleed is not secured. But it would be a simple

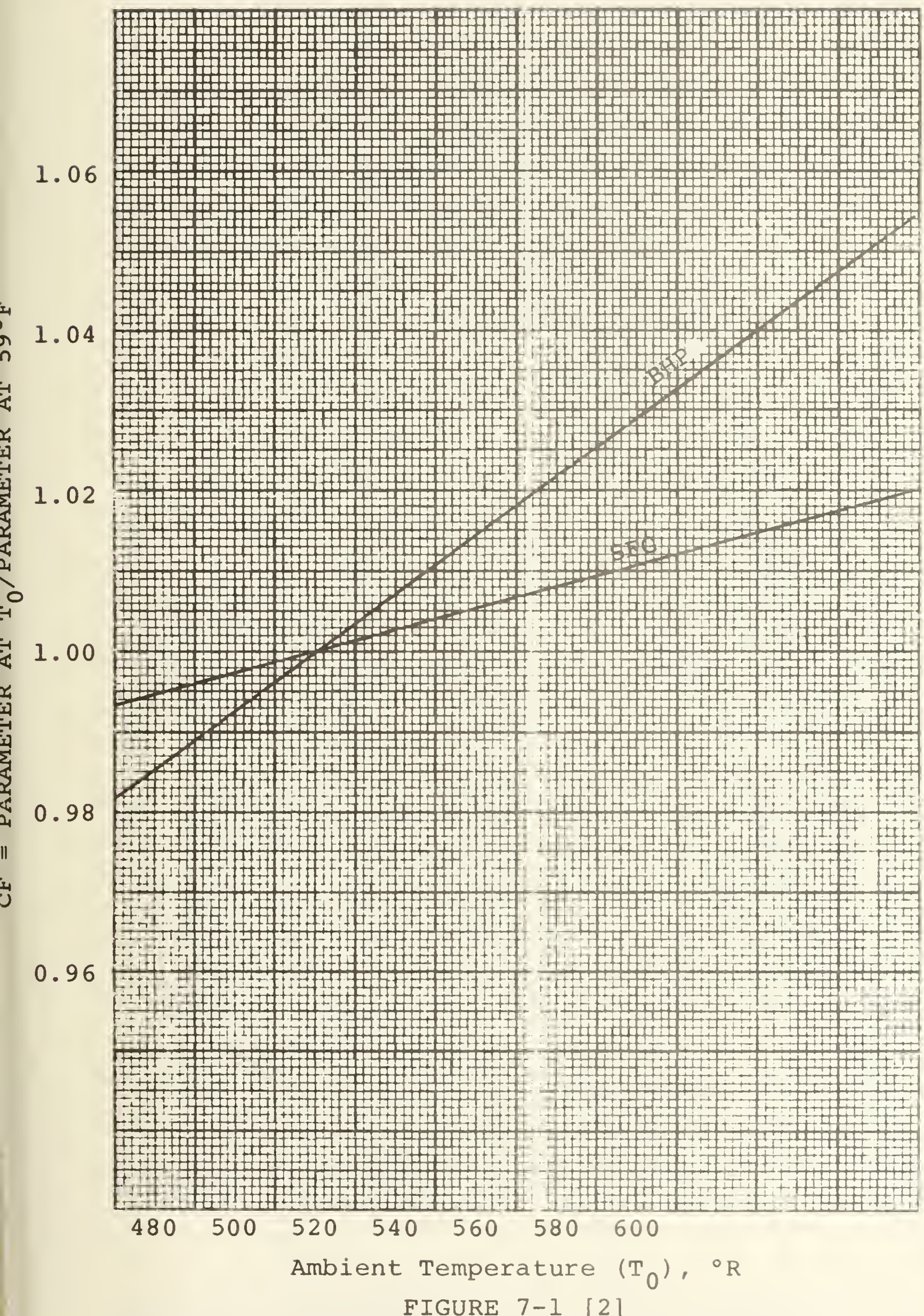

CF = PARAMETER AT T0/PARAMETER AT 59°F
1.06
1.04
1.02
1.00
0.98
0.96
BHP
SFC
480
500
520
540
560
580
600
Ambient Temperature (T0), °R

FIGURE 7-1 [2]

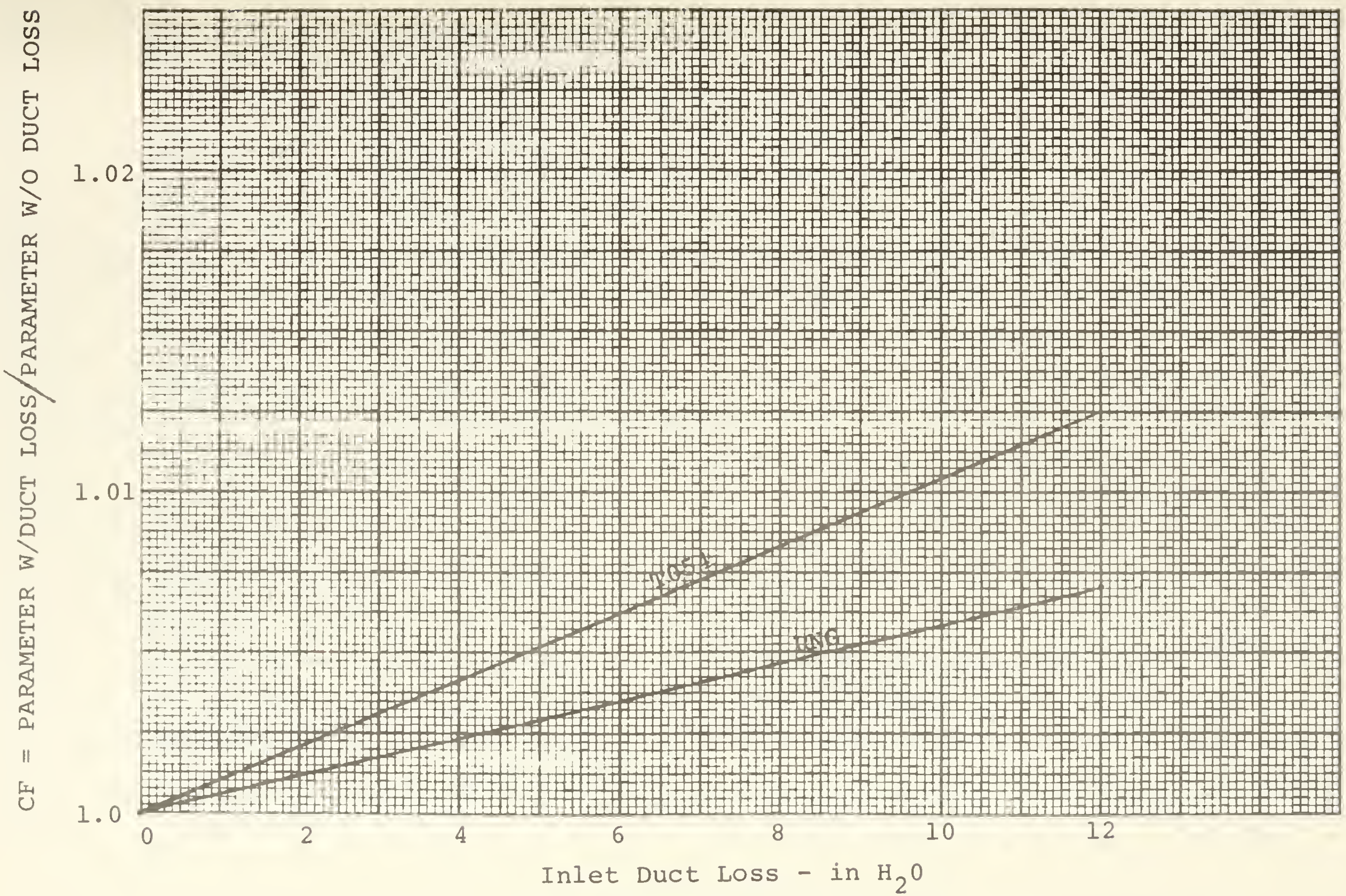

CF = PARAMETER W/DUCT LOSS/PARAMETER W/O DUCT LOSS
1.02
1.01
1.0
0
2
4
6
8
10
12
LNG
Inlet Duct Loss - in H_2O

FIGURE 7-2 [1]

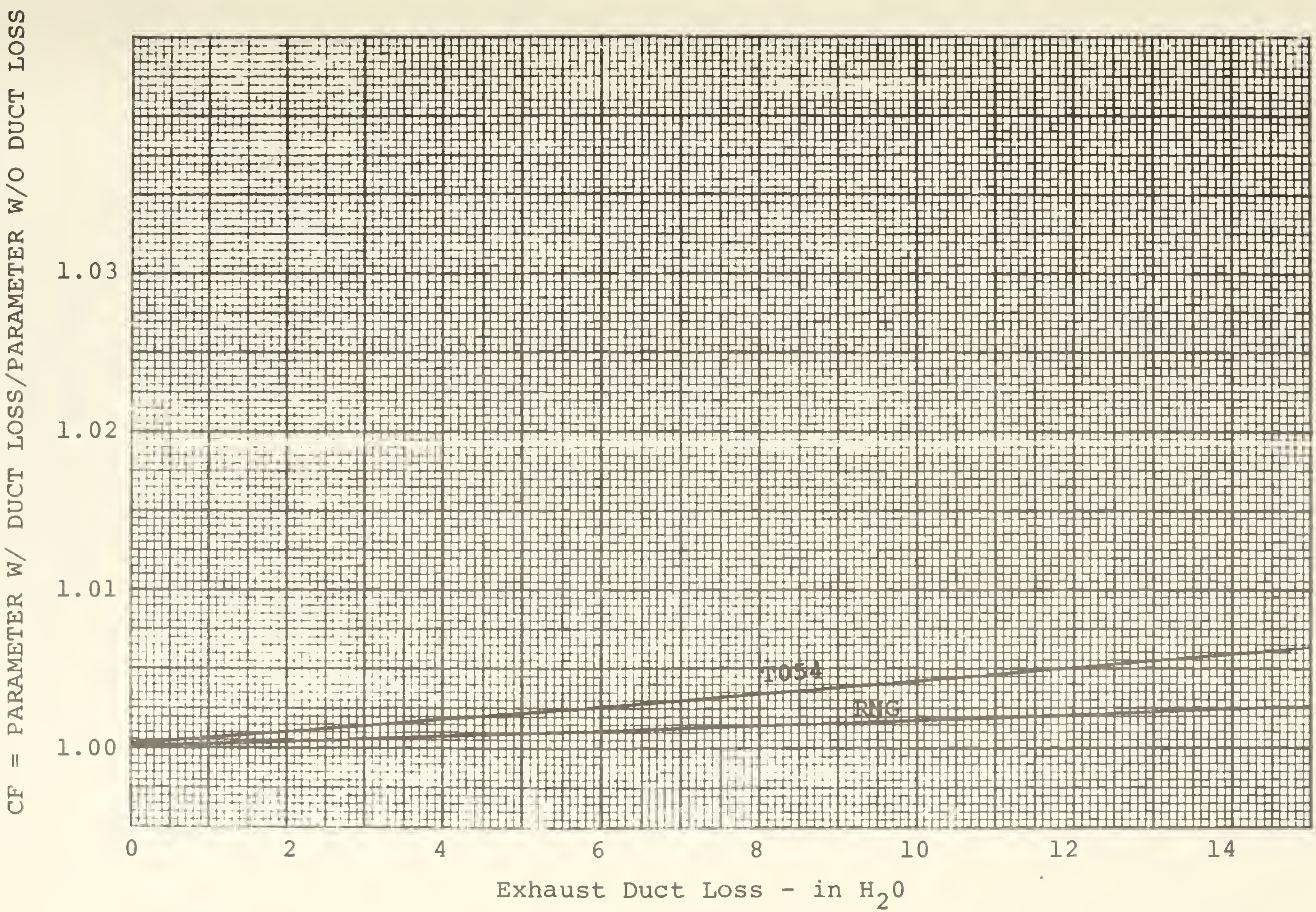

CF = PARAMETER W/ DUCT LOSS/PARAMETER W/O DUCT LOSS
1.03
1.02
1.01
1.00
0
2
4
6
8
10
12
14
T054
RNG
Exhaust Duct Loss - in H_2O

FIGURE 7-3 [1]

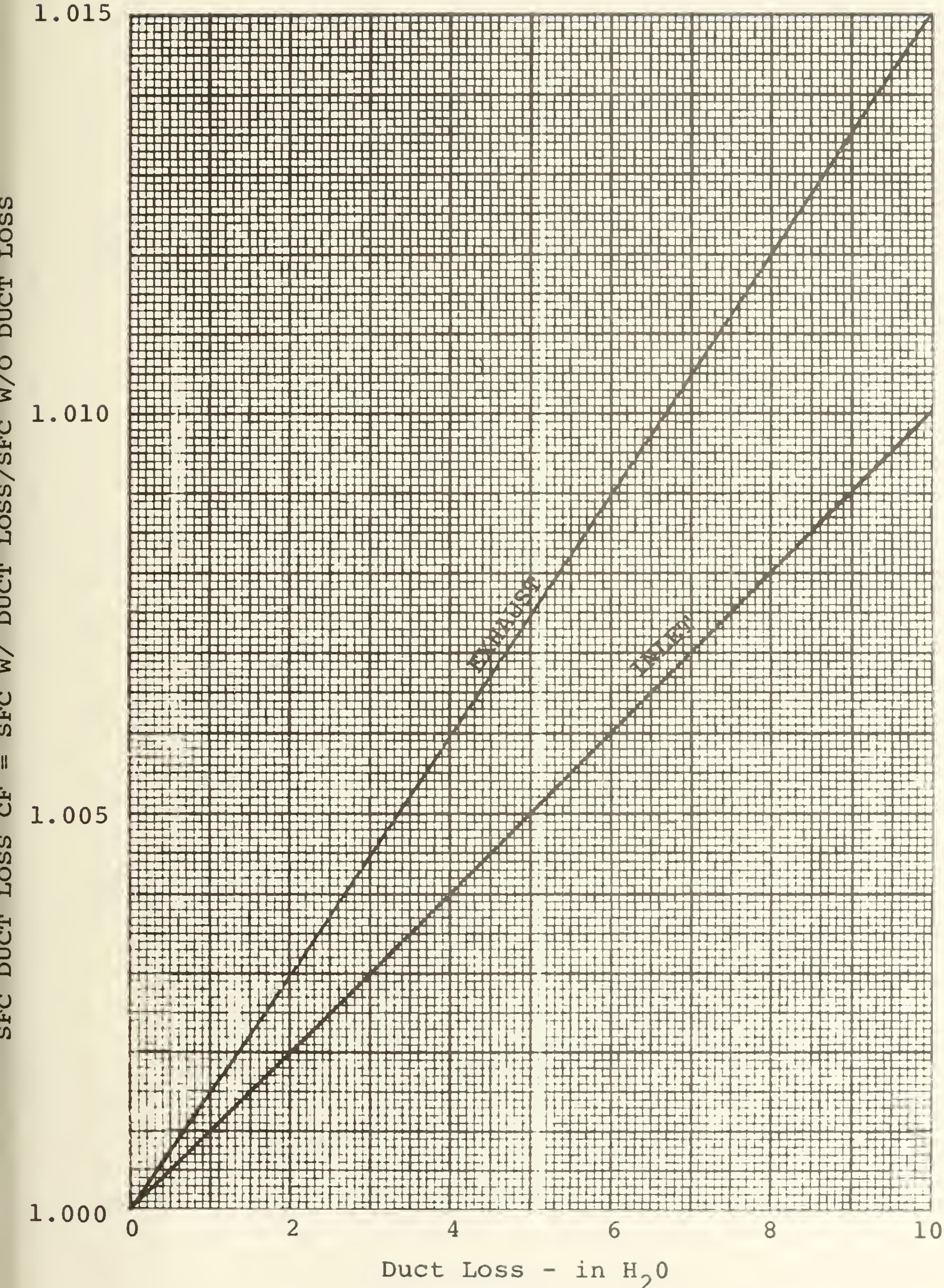

SFC DUCT LOSS CF = SFC W/ DUCT LOSS/SFC W/O DUCT LOSS
1.015
1.010
1.005
1.000
0
2
4
6
8
10
EXHAUST
INLET
Duct Loss - in H_2O

FIGURE 7-4 [2]

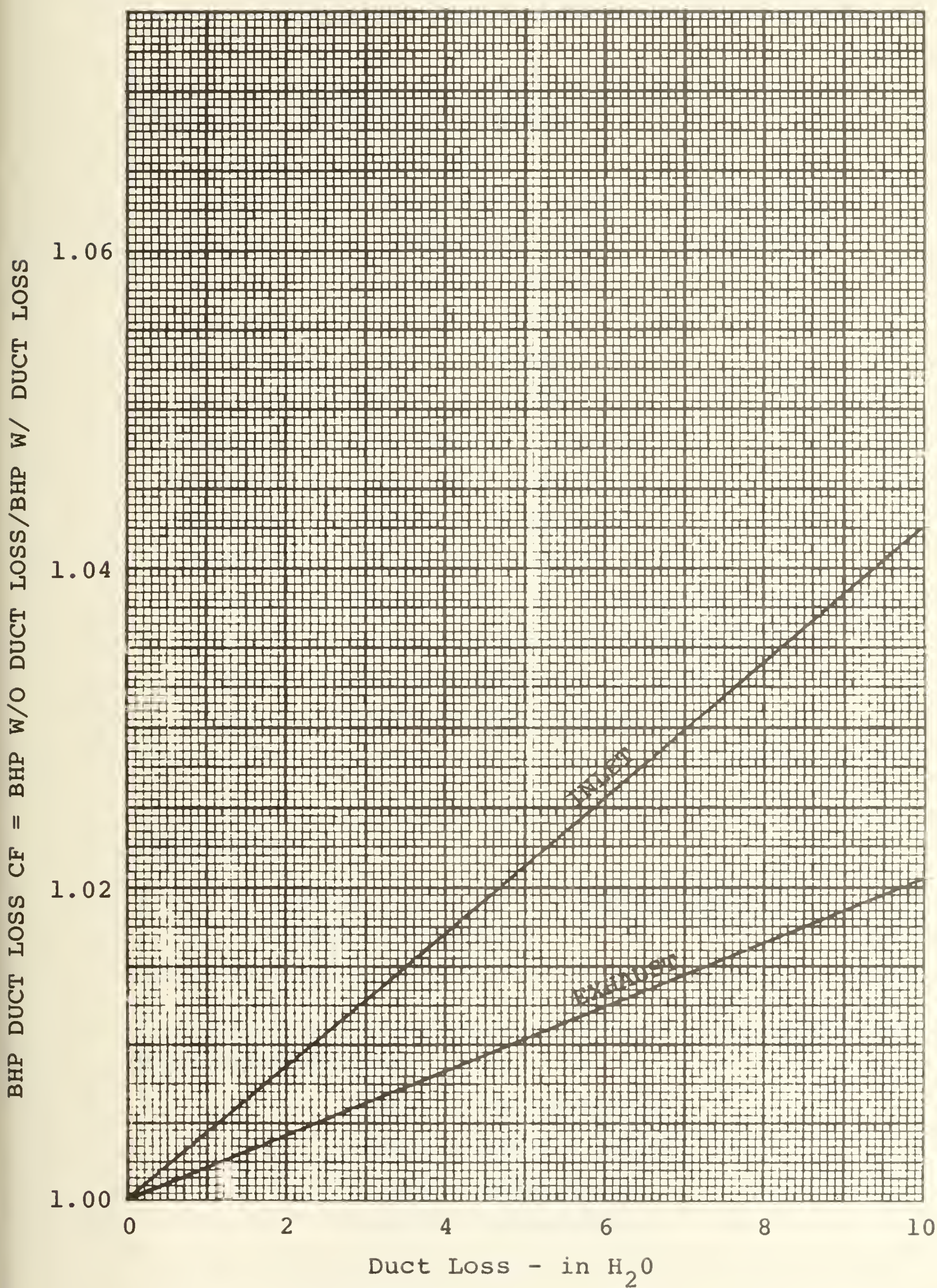

BHP DUCT LOSS CF = BHP W/O DUCT LOSS/BHP W/ DUCT LOSS
1.06
1.04
1.02
1.00
INLET
EXHAUST
0
2
4
6
8
10
Duct Loss - in H_2O

FIGURE 7-5 [2]

```
INPUT:  T054M, T02M, P02M, PS3M, WFM, RNGM,
        P054M, TORQUEM, RNPTM, INLETM, EXITM
```

```
CALCULATE CORRECTION FACTORS:

    THETA2 = T02M/518.7
   BHPCFAT = 0.0006722222 T02M + .6511166782
   SFCCFAT = 0.0002666667 T02M + .8615999827
   BHPCFIN = 0.00425 * INLETM + 1
   BHPCFEX = 0.00205 * EXITM + 1
   SFCCFIN = 0.001 * INLETM + 1
   SFCCFEX = 0.0015 * EXITM + 1
  T054CFIN = 0.00104167 * INLETM + 1
  T054CFEX = 0.0004667 * EXITM + 1
   RNGCFIN = 0.005833 * INLETM + 1
   RNGCFEX = 0.0001667 * EXITM + 1
     RNGCF = (THETA2 ** .5) * RNGCFIN * RNGCFEX
    T054CF = (THETA2**.85)*T054CFIN*T054CFEX
     SFCCF = SFCCFAT*SFCCFIN*SFCCFEX
     BHPCF = BHPCFAT*BHPCFIN*BHPCFEX
```

```
CALCULATE:

          BHPM = (RNPTM * TORQUEM)/5252
          SFCM = WFM/BHPM
```

```
APPLY CORRECTIONS:

              T054C = T054M/T054CF
               BHPC = BHPM * BHPCF
                RNG = RNGM/RNGCF
               SFCC = SFCM/SFCCF
```

FIGURE 7-6: Correction Logic

EXPLANATION OF NOTATION

M . . . . . . indicates value measured at the engine
THETA2 . . . T02M/518.7
SFCCFAT . . . ambient temperature correction factor for SFC
SFCCFIN . . . inlet duct loss correction factor for SFC
SFCCFEX . . . exit duct loss correction factor for SFC
BHPCFAT . . . ambient temperature correction factor for BHP
BHPCFIN . . . inlet duct loss correction factor for BHP
BHPCFEX . . . exit duct loss correction factor for BHP
T054CFIN . . inlet duct loss correction factor for T054
T054CFEX . . exit duct loss correction factor for T054
RNGCFIN . . . inlet duct loss correction factor for RNG
RNGCFEX . . . exit duct loss correction factor for RNG
RNGCF . . . . total correction factor for RNG
T054CF . . . total correction factor for T054
SFCCF . . . . total correction factor for SFC
BHPCF . . . . total correction factor for BHP
BHPM . . . . calculated BHP from measured parameters
SFCM . . . . calculated SFC from measured parameters
PRCM . . . . calculated PRC from measured parameters
PRGGM . . . . calculated PRGG from measured parameters
HPTPRM . . . calculated HPTPR from measured parameters
T054C . . . . T054 corrected
BHPC . . . . BHP corrected
RNGC . . . . RNG corrected
SFCC . . . . SFC corrected

matter to measure the air flows and modify the module to correct for the bleed. This would be necessary if it were to be expanded into a control unit.

The parameters are corrected to baseline with no duct losses, and the logic is shown in Figure 7-6. The correction equations are also shown in the figure.
The gas generator exit temperature is corrected for ambient temperature and also for the inlet and exhaust losses. The ambient temperature used is the compressor inlet, but it could have just as easily have been the outside air temperature. Using the compressor inlet temperature is more accurate, but in trending it is consistency that is important. The pressures are not corrected. The torque and power turbine RPM are not corrected, but the BHP is corrected. Also the fuel flow is not corrected, and the uncorrected BHP is used to calculate SFC. Then the SFC and the BHP are corrected for ambient temperature and duct losses. The procedure involves corrections for the inlet and exhaust losses and the ambient temperature.

The corrections enable the module to print out reliable data over an infinite temperature range and a horsepower range from 10000 to 25000 as was specified in Chapter VI. The most accurate data will be obtained close to 59°F, because the error introduced by the correction factors increases

slightly as the ambient temperature increases or decreases from 59°F.

CHAPTER VIII

CONCLUSIONS

The foundation of a trending program is the operational baseline. This allows the determination of deviations from normal operating conditions. The baseline can be determined for a class of engine as is one of the goals of this paper, or it can be developed directly from the operation of a new engine for that particular engine. For a trending program, either set of baseline data is acceptable because the purpose is to pick up changes from the baseline. An engine can operate, in a new condition, one percent above the class baseline for example. The trending system would detect deviations from the plus one percent point. Consequently, the general class baseline is a useful tool.

Chapter VI developed the operational baseline for the LM2500 marine gas turbine engine, and Chapter VII proposed a method of applying corrections for ambient conditions. Considering Table 6-3, each engine parameter can be expressed as a function of gas generator speed, and so the baseline is a function of the RPM. Calculating the deviation in the trending requires the measurement and calculation of the various parameters to find the actual engine operating point; and it also requires the calculation

of the corresponding parameter from the baseline for the actual, corrected gas generator speed. The deviation then is the difference. Utilizing the equations developed for the baseline in Chapter VI and the correction procedure established in Chapter VII, the equations can be assembled. The equation for the change in pressure ratio of the compressor is the difference between the actual measured pressure ratio and the baseline from the Chapter VI equation, Figure 8-1. The equation for the gas generator pressure ratio change is similar to that of the compressor and is shown in Figure 8-2. The deviation in the high pressure turbine pressure ratio is the difference between the actual value and 4.41, Figure 8-3. In finding the deviation in T054, the measured value had to be corrected and then the difference could be taken from the baseline equation of Chapter VI, Figure 8-4. The calculation of the deviations for SFC and BHP were somewhat more involved. In each case, the actual value was calculated and then corrected. Then the value of the baseline equation from Chapter VI could be subtracted. Figures 8-5 and 8-6 illustrate the calculation logic of the deviations of SFC and BHP respectively. In each case of the calculation of the baseline parameter, the RPM of the gas generator has to be corrected to standard conditions before the equation is

$$\Delta PRC = \frac{PS3M}{P02M} - \frac{\alpha * RNGM}{\sqrt{T02M} * (\delta * INLET + 1) * (\gamma * EXIT + 1)} + K$$

$$\alpha = 0.21674195$$

$$\delta = 0.005833$$

$$\gamma = 0.0001667$$

$$K = 67.43$$

FIGURE 8-1: Equation for the Deviation of PRC

$$\Delta PRGG = \frac{P054M}{P02M} - \frac{\alpha * RNGM}{\sqrt{T02M} * (\delta * INLET + 1) * (\gamma * EXIT + 1)} + K$$

$\alpha = 0.04896622$

$\delta = 0.005833$

$\gamma = 0.0001667$

$K = 15.23$

FIGURE 8-2: Equation for the Deviation of PRGG

$$\Delta HPTPR = \frac{PS3M}{P054M} - 4.41$$

FIGURE 8-3: Equation for the Deviation of HPTPR

$$\Delta T054 = \frac{\alpha * T054M}{T02M ** .85 * (a*INLET + 1) * (b*EXIT + 1)} - \frac{\beta * RNGM}{\sqrt{T02M} * (\delta*INLET + 1) * (\gamma*EXIT + 1)}$$

$$+ \frac{\Gamma * RNGM ** 2}{T02M * (\delta*INLET + 1) ** 2 * (\gamma*EXIT + 1) ** 2} + K$$

$\alpha = 203.08545$	$\delta = 0.005833$
$a = 0.00104167$	$\gamma = 0.0001667$
$b = 0.0004667$	$\Gamma = 0.22346829$
$\beta = 181.71037$	$K = 34993.88$

FIGURE 8-4: Equation for the Deviation of T054

$$\Delta SFC = \frac{\alpha * WFM}{RNPT * TORQUEM * (a*T02M + b) * (c*INLET + 1) * (d*EXIT + 1)}$$

$$- \frac{\beta * RNGM}{\sqrt{T02M} * (\delta*INLET + 1) * (\gamma*EXIT + 1)} - \frac{\Gamma * RNGM ** 2}{T02M * (\delta*INLET + 1) ** 2 * (\gamma*EXIT + 1) ** 2} - K$$

$\alpha = 5252$ $\beta = 0.10380365$

$a = 0.0002666667$ $\delta = 0.005833$

$b = 0.86159998$ $\gamma = 0.0001667$

$c = 0.001$ $\Gamma = 0.00013452889$

$d = 0.0015$ $K = 20.417603$

FIGURE 8-5: Equation for the Deviation of SFC

$$\Delta BHP = RNPTM * TORQUEM * \beta * (a*T02M + d) * (b*INLET + 1) * (c*EXIT + 1)$$

$$- \frac{\alpha * RNGM}{\sqrt{T02M} * (\delta*INLET + 1) * (\gamma*EXIT + 1)} + K$$

$\beta = 0.0001904037$ $\alpha = 569.37466$

$a = 0.0006722222$ $\delta = 0.005833$

$d = 0.65111668$ $\gamma = 0.0001667$

$b = 0.00425$ $K = 195000$

$c = 0.00205$

FIGURE 8-6: Equation for the Deviation of BHP

entered. This is reflected in the equation. Each of the equations is expressed as only functions of measured parameters, and the correction factors are incorporated. They are the trending system when coupled with engine hours of operation. The equations are useful over the horsepower range previously specified and over an infinite temperature range. The only limiting factors are that the engine must be in a steady cruising state* and that ship service bleed is secured.

The trending can be accomplished by several means. The equations could be applied to a software program in which the engine log data would be the input. A computer facility ashore could be a central trending center for all of the LM2500 ships. Another possible method would be to program a hand calculator to be supplied to the engineer. A further use of the equations would be to build them into an analog circuit. However, to achieve a desired accuracy, cost would be prohibitive. The most advantageous use of the equations is to incorporate them into an analog-digital computer. The cost would be less than the analog board, and the desired accuracy would be attained. Such a device,

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* Defined as attained when the power setting has not been altered in five minutes [5].

when installed on the engine, would not require trained technicians or its own power supply. Since the cost could be kept down, the unit could be treated as a black box, and repair would consist of drawing a new computer from supply and installing it. Having the system on board makes the trending information immediately available to the ship's personnel, who are immediately responsible to meet the commitments. This is the disadvantage of a shore based software program. The analog-digital computer would be the most effective use of the equations in the marine environment.

As an extension of the analog-digital computer, the system could be adapted for control and on-condition monitoring. This would require some modifications to the logic, but the information which would be required was included in the data reduction in Chapter VI. The use of the computer module as a monitor/controller would also simplify the conning of the ship.

The total control system could consist of two similar modules. One module would control the engine for cruising. It would be programmed for the most economical mode of operation and would involve dynamics only in transitting from one speed to another. It could be designated "mode-one" and would operate over a speed range from zero

to full power. The control lever would be calibrated in knots. Thus, the conning officer would request "mode one" and specify RPM and knots. "Mode-two" would correspond to another module which would be programmed to yield the most responsive maneuvering characteristics. This "mode" would also operate over a speed range of zero to full power and be utilized in maneuvering situations such as entering or leaving port and steaming alongside or in formation. The conning officer would give engine commands by specifying the mode and ordering a power level. He would then control the ship by changing the pitch. This method would allow for a reserve of torque. The level of power would correspond to a particular power turbine speed, and as the torque reserve is used up, the speed would change corresponding to a new level. There would be several control levels and each control level would have a range of pitch. This control proposal would eliminate the conning officer having to adjust around the present twelve knot changeover, where the commands go from degrees of pitch to RPM's, and allow for more positive control.

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APPENDIX A

NOMENCLATURE

Name	Symbol	Unit
Ambient Temperature	T0	Deg. R
Ambient Pressure	P0	PSIA
Compressor Inlet Total Temperature	T02	Deg. R
Compressor Inlet Total Pressure	P02	PSIA
Power Turbine Speed	RNPT	RPM
Engine Brake Horsepower (Note 1)	BHP	HP
Specific Fuel Consumption (Note 2)	SFC	Lb/HP-Hr
Heat Rate	HR	BTU/HP-Hr
Compressor Inlet Air Flow	W2	Lb/Sec
Physical Gas Generator Speed	RNG	RPM
Compressor 16th Stage Bleed Flow Rate	WB16	Lb/Sec
Compressor 16th Stage Bleed Pressure	PE16D	PSIA
Compressor 16th Stage Bleed Temperature	TE16D	Deg. R
Fuel Flow	WF	Lb/Hr
Lower Heating Value of Fuel	LHV	BTU/Lb
Power Turbine Inlet Total Temperature	T054	Deg. R

Name	Symbol	Unit
Power Turbine Inlet Total Pressure	P054	PSIA
Pressure Ratio of Gas Generator (P054/P02)	PRGG	----
Power Turbine Shaft Torque	Torque	Ft-Lbs
Exhaust Duct Discharge Total Pressure	P08	PSIA
Exhaust Duct Discharge Total Temperature	T08	°R
Exhaust Duct Discharge Flow	W8	Lb/Sec
Exhaust Duct Discharge Specific Heat	CP8	BTU/Lb-Deg. R
Inlet Loss (ship's inlet air induction system pressure drop)	INLET	In. of water
Exhaust Loss (ship's exhaust ducting system backpressure)	EXIT	In. of water

APPENDIX B

ESTIMATED AVERAGE PERFORMANCE OF THE LM2500 ENGINE AT 59°F [1]

INLET	= 0
EXIT	= 0
AUXILLIARY BLEED	= 0
ALTITUDE	= 0
HUMIDITY	= 0

ESTIMATED AVERAGE PERFORMANCE

BHP = 10000

T0	518.7	518.7	518.7	518.7	518.7	518.7
P0	14.696	14.696	14.696	14.696	14.696	14.696
T02	518.7	518.7	518.7	518.7	518.7	518.7
P02	14.696	14.696	14.696	14.696	14.696	14.696
RNPT	1200.	1800.	2400.	3000.	3300.	3600.
BHP	10000.	10000.	10000.	10000.	10000.	10000.
SFC	0.6865	0.5307	0.4827	0.4889	0.5014	0.5181
HR	12631.	9765.	8881.	8995.	9226.	9534.
W2	120.8	105.6	99.2	97.1	97.1	97.5
RNG	8500.4	8262.6	8164.3	8133.2	8133.6	8139.7
PE16D	194.9	163.2	151.2	149.1	150.1	151.8
TE16D	1176.	1119.	1093.	1089.	1091.	1095.
WF	6865.	5307.	4827.	4889.	5014.	5181.
LHV	18400.	18400.	18400.	18400.	18400.	18400.
T054	1628.	1510.	1480.	1512.	1536.	1563.
P054	42.45	36.86	35.48	36.34	37.13	38.07
PRGG	2.889	2.508	2.414	2.473	2.526	2.590
TORQUE	43767.	29178.	21884.	17507.	15915.	14589.
P08	14.78	14.75	14.75	14.74	14.74	14.75
T08	1407.	1253.	1205.	1233.	1258.	1287.
W8	121.6	106.1	99.6	97.6	97.6	98.0
CP8	0.2719	0.2654	0.2635	0.2647	0.2658	0.2671

ESTIMATED AVERAGE PERFORMANCE

BHP = 15000

T0	518.7	518.7	518.7	518.7	518.7
P0	14.696	14.696	14.696	14.696	14.696
T02	518.7	518.7	518.7	518.7	518.7
P02	14.696	14.696	14.696	14.696	14.696
RNPT	1800.	2400.	3000.	3300.	3600.
BHP	15000.	15000.	15000.	15000.	15000.
SFC	0.5151	0.4470	0.4337	0.4374	0.4467
HR	9478.	8224.	7980.	8048.	8219.
W2	126.9	117.0	112.8	111.8	111.7
RNG	8604.9	8451.7	8380.9	8365.4	8363.3
PE16D	209.6	188.6	181.6	180.7	181.4
TE16D	1201.	1166.	1154.	1153.	1155.
WF	7726.	6705.	6506.	6561.	6700.
LHV	18400.	18400.	18400.	18400.	18400.
T054	1699.	1637.	1643.	1661.	1684.
P054	45.83	42.53	42.40	42.91	43.74
PRGG	3.119	2.894	2.885	2.920	2.976
TORQUE	43767.	32826.	26261.	23873.	21884.
P08	14.79	14.77	14.77	14.76	14.77
T08	1387.	1296.	1290.	1305.	1329.
W8	127.9	117.8	113.6	112.6	112.5
CP8	0.2720	0.2681	0.2680	0.2687	0.2698

ESTIMATED AVERAGE PERFORMANCE

BHP = 20000

T0	518.7	518.7	518.7	518.7
P0	14.696	14.696	14.696	14.696
T02	518.7	518.7	518.7	518.7
P02	14.696	14.696	14.696	14.696
RNPT	2400.	3000.	3300.	3600.
BHP	20000.	20000.	20000.	20000.
SFC	0.4317	0.406[illegible]	0.4062	0.4088
HR	7944.	7485.	7475.	7522.
W2	132.9	126.9	125.4	124.5
RNG	8722.9	8606.9	8583.1	8567.9
PE16D	224.3	212.3	210.2	209.2
TE16D	1225.	1206.	1203.	1201.
WF	8634.	8136.	8125.	8176.
LHV	18400.	18400.	18400.	18400.
T054	1769.	1755.	1767.	1782.
P054	49.50	48.28	48.58	49.12
PRGG	3.368	3.285	3.305	3.343
TORQUE	43768.	35014.	31831.	29178.
P08	14.79	14.78	14.78	14.78
T08	1377.	1342.	1350.	1363.
W8	134.1	128.0	126.5	125.6
CP8	0.2724	0.2710	0.2714	0.2720

ESTIMATED AVERAGE PERFORMANCE

BHP = 25000

T0	518.7	518.7	518.7
P0	14.696	14.696	14.696
T02	518.7	518.7	518.7
P02	14.696	14.696	14.696
RNPT	3000.	3300.	3600.
BHP	25000.	25000.	25000.
SFC	0.3877	0.3870	0.3872
HR	7134.	7120.	7125.
W2	140.6	138.6	136.8
RNG	8853.7	8799.9	8775.2
PE16D	243.0	239.6	236.9
TE16D	1250.	1245.	1243.
WF	9693.	9675.	9680.
LHV	18400.	18400.	18400.
T054	1836.	1851.	1867.
P054	54.12	54.30	54.56
PRGG	3.683	3.695	3.713
TORQUE	43768.	39789.	36473.
P08	14.81	14.81	14.81
T08	1377.	1386.	1397.
W8	142.1	140.1	138.3
CP8	0.2731	0.2736	0.2741

Thesis
T345
c.1
Tetlow
184500
A thermodynamic study of a representative gas turbine for design of a computer module for steady state engine monitoring and trending.